Propulsion and Vehicle Engineering Division
Propulsion and Mechanics Branch

Final Report
EVALUATION OF HEAT-FLUX METERS
Phase 2 - Experimental Investigation

MARSHALL SPACE FLIGHT CENTER

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Final Report

EVALUATION OF HEAT-FLUX METERS

Phase 2 - Experimental Investigation

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SUMMARY

A

An experimental program to evaluate the accuracy and applicability of slug-type heat-flux meters has been conducted. The aim of the program was to substantiate the conclusions drawn under Phase 1, an analytical study, and to further determine the major design parameters involved in the use of slug-type heat-flux meters.

An additional objective of the program was to establish a practical and versatile laboratory procedure that would provide convective and radiant heating separately and concurrently for laboratory calibration of heat-flux meters. An apparatus having this capability was designed and constructed. It consists of a 30-tip oxyacetylene torch and an electrically heated graphite-block radiator with a 6" × 6" heating surface. Standardization of heat flux from the apparatus was established by means of precise temperature measurements on a special thin plate of stainless steel for which the heat capacity was precisely known.

Two meter designs were tested. In the Type A meter, no attempt was made to match the thermal characteristics of the meter and the structure in which it was heated. The second design (Type B) provided better insulation for the slug and an improved matching to the mounting structure.

The results of tests conducted on these meters indicated that when they are heated by a convective source, their behavior differs from that when they are heated by an equivalent flux produced by a radiation source.

Limited testing of Type A meters indicated errors of approximately 60%, while maximum errors of 25% were observed in the flux indicated by the Type B meters. This magnitude of error has been attributed primarily to various perturbations caused by the design of the meter assembly tested.

Author

INTRODUCTION

In the study of high heat transfer rates, particularly those associated with aerodynamic re-entry of orbital or ballistic vehicles and rocket-engine work, considerable use has been made of heat-flux meters of various designs. Phase 1 of this program was an analytical investigation of the accuracy and applicability of several of these meters, with emphasis being placed primarily on single-capacity or slug-type meters because of their convenience and widespread use. Experience had pointed out that results obtained with these meters were at times not in full agreement with theoretical predictions, the reasons for this disagreement not being apparent. A study of the important parameters that affect the accuracy of these meters was therefore initiated.

It was concluded under Phase 1[1] that errors of 20% or more are probably not uncommon and that larger errors might exist, depending on the care taken in the meter design and installation. In addition, it was concluded that reliable laboratory calibration may be difficult to achieve, since the performance of a meter may be dependent upon the heating mode encountered (i.e., convection or radiation). Thus the calibration for a meter obtained under radiant-heating conditions may not be strictly applicable for the meter when used to measure convective heating.

The current program, Phase 2, was aimed at the experimental verification of these conclusions and the evaluation under various heating conditions of slug-type meters. Because of the large variety of meter designs possible, not all of them could be tested, but some of the parameters important in these designs were evaluated.

The text of this report contains a general description of the experimental work done and the important results obtained. Detailed descriptions of the analytical methods employed appear in Appendix A, and a summary of the experimental data obtained can be found in Appendix C.

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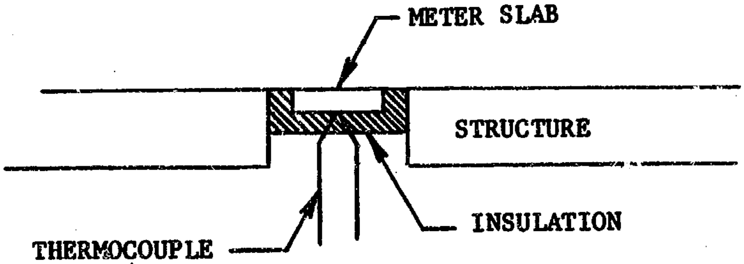
1. "Analytical Investigation of Heat-Flux Meters," Advanced Technology Laboratories, a Division of American-Standard, Final Report, ATL-D-711, 31 October 1961.

SLUG-TYPE HEAT-FLUX METER

A. Theory of Operation

A slug-type heat-flux meter is usually a thin slab of high-thermal-conductivity material, thermally isolated from its surroundings, and having a thermocouple attached to its unheated surface. It is basically a single-thermal-capacitance system whose temperature and rate-of-change of temperature are a measure of the total imposed heating and the instantaneous heating rate, respectively. The following sketch shows the salient features.



The heat content of an isothermal mass is given by:

$$Q_t = \int_0^T \rho c_p V dT \ , \tag{1}$$

where T = temperature,

ρ = density,

c_p = specific heat,

V = volume per unit surface area.

For the case of constant thermal properties, this expression reduces to

$$Q_t = \rho c_p VT \ . \tag{2}$$

The rate of heating is then given by the time rate-of-change of the heat content:

$$\dot{Q} = \frac{dQ_t}{dt} = \rho c_p V \frac{dT}{dt} \ , \tag{3}$$

where t = time.

Hence, the heating rate can be determined by measuring the temperature history of the isothermal mass and then determining its rate-of-change graphically or by electronic computer. In an actual system, the slug has a finite thermal conductivity; hence the assumption must be made that the temperature measured is the average temperature, or at least that its time derivative is the same as that of the average temperature. For thin slugs of high-conductivity material such as those used in this test program (i.e., 1/8- and 1/4-inch-thick copper), this assumption is a valid one.

B. Factors Contributing to Errors

The analysis carried out in Phase 1 of this program indicated that several factors are important when considering the errors that might be expected in the use of slug-type heat-flux meters.* It was concluded that disturbance of the thermal path is probably the most important factor. A meter that assumes a temperature substantially different from that of the surrounding structure is susceptible to heat losses or gains from the surrounding structure. Furthermore, large temperature differences may alter the convective heating by changing the thermal boundary layer, and thus in turn changing the convective heat transfer coefficient in the vicinity of the meter. This factor may be the major reason that there is a difference between the errors to be expected under convective and radiant heating as measured with a given meter.

Other errors may exist because of time-response characteristics, rear-surface heat losses, and thermocouple-fin heat losses (i.e., conduction along the thermocouple leads which degrades the recorded temperature). However, none of these has as serious an effect as the disturbance of the thermal path.

The above considerations indicated that in order to check the validity of these conclusions experimentally, it is necessary to provide controlled heating by both convection and radiation. With this in mind, the construction of a laboratory heating apparatus with provisions for convection, radiation, and combined convection-radiation heating was undertaken.

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* See Phase 1 final report (reference 1) for a complete analysis of these factors.

HEATING APPARATUS

The heating apparatus constructed consists essentially of a 30-tip oxyacetylene heating torch and an electrically heated graphite block. The heating-torch system includes seven manifolded acetylene bottles and a single oxygen bottle, a flowmeter for each gas to insure close reproducible control of heating rates, solenoid valves for automatic operation, and throttling valves for fine control of gas flow. The graphite block is heated by current from a power supply capable of developing 75 kilowatts. This block is bathed in argon during the heat-up period, and a shutter is opened automatically in conjunction with the opening of the gas solenoid valves. The block temperature is monitored and controlled with a Leeds & Northrup Rayotub and power controller.

Figures 1 and 2 (front and rear view, respectively) show the over-all layout of t heating apparatus.* The central portion of Figure 1 shows the graphite radiant heat with water-cooled cover and control Rayotube directly above it. The large slotted st provides variable positioning of the torch, which in this photograph is shown in a hor zontal position. Positions of the torch other than horizontal are for convection heati only, without the radiation source. On the right is the water-cooled exhaust duct; or the left is a portion of the instrument panel containing various control switches, thro valves, and flowmeters.

Figures 3 and 4 are close-up views of the heating zone. In Figure 4, the water-cooled cover has been removed from the graphite block. A 0.125-inch-thick stainle steel test plate (described below) is shown in place, recessed into the brick base. I warmup, the under side of the radiant heater is covered by a sliding shutter that is actuated by an air cylinder. This shutter is visible in Figure 3. Figure 5 is a sche of the apparatus.

The entire test procedure is automated. The oxygen and acetylene solenoid valv and a solenoid valve controlling air to a pencil-type cylinder driving the shutter are operated by a microswitch sequence timer. The oxygen valve opens first, followed

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* All illustrations appear following page 20 in a section immediately preceding the appendices.

closely by the acetylene valve. A pilot flame, which is ignited prior to initiation of the timing sequence, ignites the torch at the instant the acetylene valve opens. When both convective and radiant heating are to be used, the graphite heater is brought to equilibrium before the timing sequence is initiated. Equilibrium is determined visually with an optical pyrometer looking through a hole in the heater cover. The solenoid valve controlling the shutter-actuating air cylinder is opened by the timer in conjunction with the acetylene valve. The result is essentially a step heat input. The test then proceeds for a predetermined period of time, after which a reverse shut-off sequence occurs.

EXPERIMENTAL PROCEDURE

A. Standard Flux Measurements

Following is a brief description of the procedure employed in establishing a known heat flux and the basic theory supporting the procedure.

Convective heating rate to a surface is given by:

$$\dot{Q}_c = h(T_g - T_s) \quad . \tag{4}$$

The heat transfer coefficient, h, is dependent upon mass flow rate and gas-stream properties. It can be assumed that h is a constant by maintaining the mass flow rate of both the oxygen and the acetylene constant. Maintaining the gas flows constant will also permit the assumption of constant gas temperature, T_g, since the combustion conditions are invariant. The only remaining variable on the right-hand side of equation 4 is thus the surface temperature, T_s. The validity of these assumptions is assured by measuring oxygen and acetylene flow with precision flowmeters. These meters have a 10-inch scale with 100 equal divisions. Each gas line is provided with a throttling valve for fine adjustment so that the flow can be easily reproduced within 1% of a specified value.

Radiant heating rate to a surface is given by:

$$\dot{Q}_r = \sigma \alpha_s (\epsilon_r F T_r^4 - T_s^4) \quad . \tag{5}$$

The geometrical-shape factor, F, is a constant of the system, and σ is the Stefan-Boltzmann constant. The surface absorptivity of the plate surface, α_s, and the emissivity of the graphite plate, ϵ_r, will both be about 0.9 and can be assumed to be constant, since the test plate and meters are prepared with a special high-absorptivity high-temperature finish. The radiant-source temperature, T_r, is controlled through a feedback to the power controller from a Rayotube viewing the source. T_r can therefore also be assumed constant. The only remaining variable on the right-hand side of equation 5 is, as in equation 4, the surface temperature, T_s.

The total heat transfer to a surface subject to both convective and radiant heating is given by:

$$\dot{Q} = \dot{Q}_c + \dot{Q}_r \tag{6}$$

Since, under any constant set of test conditions to be employed, equations 4 and 5 were shown to be functions of T_s only, $\dot{Q}$ in equation 6 (the heating rate to the surface) will also be a function of the surface temperature only. This fact provides the basis for the method of evaluation.

The first step is to determine the true heating rate to a test plate under a given set of mass-flow and radiant-heating conditions. This is accomplished by determining the heat stored in a thin standardizing plate as a function of time. The slope of the resulting curve of heat stored versus time is the rate of heat storage. This rate of storage plus the rate of heat loss from the rear surface of the plate is the true heating rate.

In order to determine the total heat content of the standardizing plate at all times, the average temperature of the plate must be known. Thermocouples are located at the surface, midplane, and rear of the plate, and it is assumed that each measured temperature is the average temperature of the zone of the plate in which the thermocouple is located. Records of the variation in the three temperatures with time, combined with the heat content of the plate material (known at all temperatures from room temperature to 2000°F), permit accurate calculation of the heat stored in the standardizing plate as a function of time. Appendix A gives the computation details for rear-surface losses.

The material of the standardizing plate, Type 316 stainless steel, chosen for its non-magnetic, oxidation-resistant, and machining properties, was submitted to the University of California, Berkeley, for accurate heat-content (enthalpy) measurements as a function of temperature. The results of these measurements are shown in Appendix B.

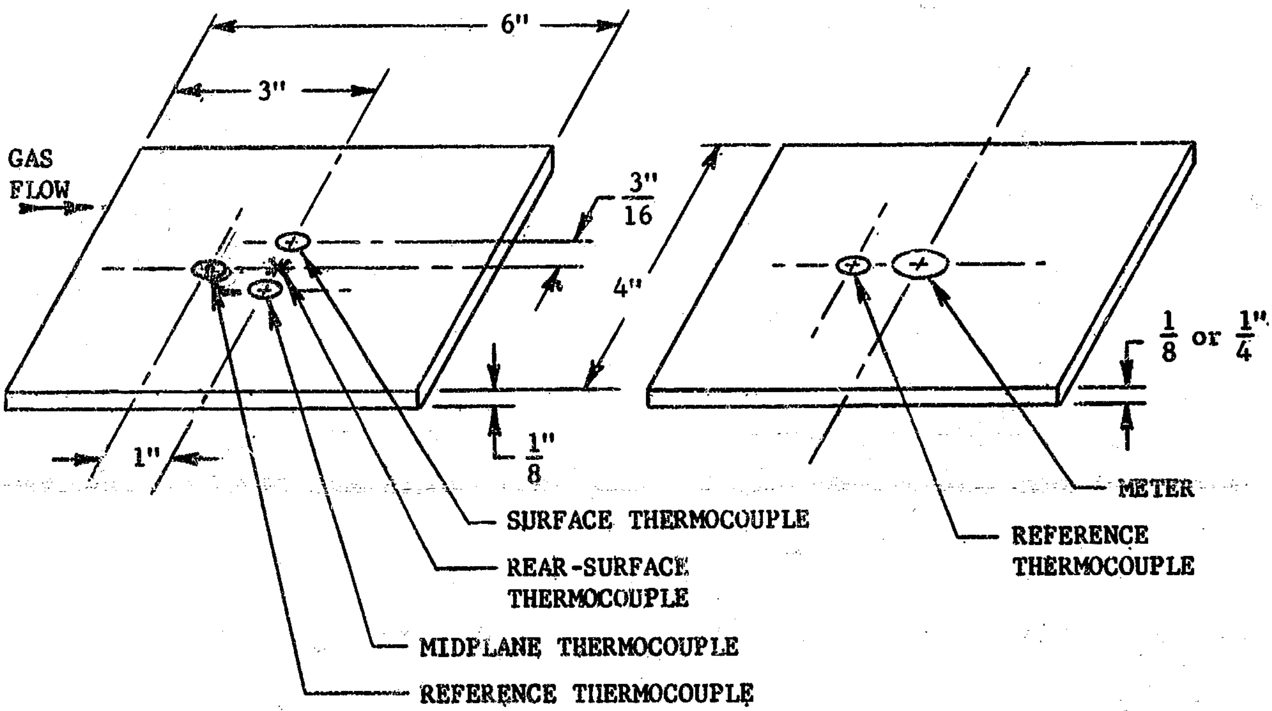
Once the true heating rate has been determined by the above method for a given set of power and gas-flow settings, it is known and can be plotted as a function of surface temperature. All subsequent tests are then correlated with the true heating rate, the surface temperatures being the basis for comparison.

B. Data Control

The methods used to calibrate heat-flux meters require the assumption that the heating conditions are identically reproduced from test to test. In order to assure reproducibility, a reference surface temperature measurement was made 1 inch upstream from the meter location, corresponding in location to an identical surface temperature measurement made on the standardizing plate. Even though the meter may have altered the thermal characteristics of the plate, the reference thermocouple was far enough away that it was not affected. Thus, the heating rates during sequential tests with plates of the same thickness were reproduced if the reference temperature histories agreed. In general, reproducibility was excellent.

C. Test Plates and Meters

The configurations of typical test plates are shown in the sketch below. The plates are, as previously noted, Type 316 stainless steel. All plates and meters were painted with a high-temperature black paint.



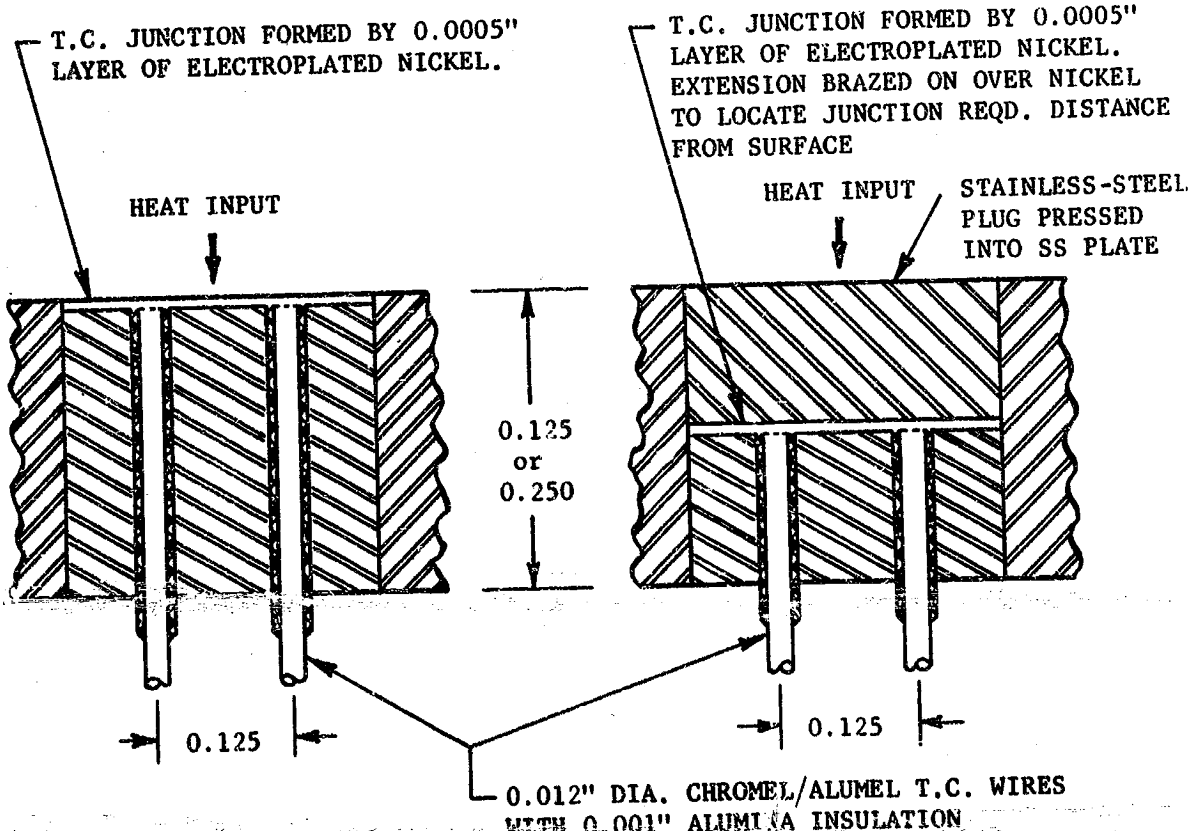
Standardizing Plate

Meter Test Plate

Surface and in-wall temperature sensors were made using techniques well established in the manufacture of standard ATL temperature sensors (Delta-Couples). See sketch below. These sensors are made by locating the junction of the thermocouple at the desired depth from the heated surface in a plug of the test-plate material. The instrumented plugs are then pressed into the test plate, providing a homogeneous system with a minimum of thermal disturbance due to the presence of the sensors.

a. Surface Thermocouple

b. Midplane Thermocouple



EMBEDDED THERMOCOUPLES (DELTA-COUPLES, PROPRIETARY AMERICAN-STANDARD PRODUCT) INCORPORATING PRECISE LOCATION OF JUNCTION BY PLATING AND BRAZING TECHNIQUES

Two separate heat-flux-meter designs were used. Type A meters consist of a copper slug in an alumina insert, which is in turn held in a stainless-steel ring pressed into the test plate. Figure 6 shows an assembly of this meter. The slug thickness can be changed by replacing both the slug and the alumina insert. The stainless-steel ring size is constant at 5/16 inch thick.

Type B meters consist of a copper slug with a 1/16-inch-thick layer of zirconia flame-sprayed on its periphery. The zirconia is then cemented directly into the test plate. The Type B meter is shown in Figure 7. Both meters use 0.005-inch-diameter wire for the thermocouple spot-welded to the rear surface. This small wire reduces fin losses to a minimum.

D. Data Acquisition

The signal from each thermocouple in a test specimen was amplified and recorded on a Minneapolis-Honeywell "Visicorder." The Visicorder was calibrated just prior to each test. A Leeds & Northrup Model 8662 potentiometer was used as a voltage reference.

DATA REDUCTION

In order to determine heating rate, the temperature records (except for the reference temperature) were converted to a plot of heat content versus time, using the heat-content data obtained at the University of California (ref. Appendix B), for the zone of the standardizing plate in which the thermocouple was located. It was assumed that the surface thermocouple recorded the average temperature of one-fourth of the plate (i.e., a 0.031-inch-thick zone), the midplane thermocouple the average temperature of one-half of the plate, and the rear-surface thermocouple the average temperature of the remaining one-fourth of the plate. The slope of the heat-content curves (measured graphically) then represented the heating rate to each zone. The sum of these three heating rates and the rear-surface conduction losses was the desired heating rate for the plate. The rear-surface losses are small (a maximum of about 3%) but should be considered (see Appendix A).

Data for the slug-type meters were reduced in the same fashion, except that the measured temperature was assumed to be the average temperature of the entire slug. The heat content for copper was determined from specific heat data in the literature.[2,3] The specific heat was assumed to be given by

$$C_p = 0.092 + 1.142 \times 10^{-5}\, T \quad \text{Btu/lb-°F} . \tag{7}$$

The heat content as a function of temperature is then given by

$$Q_t = \int_0^T C_p dT = 0.092T + 0.571T^2 \times 10^{-5} \quad \text{Btu/lb} . \tag{8}$$

2. C. F. Lucks and H. W. Deem, "Thermal Properties of Thirteen Metals," ASTM Special Technical Publication No. 227, February 1958.
3. A. Goldsmith, T. E. Waterman, and H. J. Hirschborn, Handbook of Thermo-physical Properties of Solid Materials, Revised Edition, The MacMillan Co. (1961).

RESULTS

A. Type A Meters

The first series of tests was run using the Type A meters. The majority of the tests were with convective heating, with a limited number of combined convection-radiation tests. Figures 8, 9, and 10 show the most significant results of this series. Reference temperatures used as a data control to show reproducibility are tabulated in Appendix C. These show an average spread in temperature of ± 4% from test to test.

The design of the Type A meter was not intended to match the thermal characteristics of the test plates, but rather was intended to demonstrate the magnitude of errors which can result if inadequate insulation is provided between the slug and the surrounding structure. Results for this meter assembly (see Figures 8, 9. and 10) demonstrate that this design will not indicate heat flux accurately, the flux indicated by the 1/8-inch-thick meter, for example, being in disagreement with the true flux by as much as 60%. When the 1/8-inch meter is used in a plate of the same thickness. the error is slightly less than when it is used in a 1/4-inch-thick plate. The accuracy of the meter with a 1/4-inch slug is considerably better because less heat is exchanged between the slug and the holder ring. but it is still in error by as much as 30%.

B. Type B Meters

In view of the large errors observed in the measurements made with the Type A meters, and the fact that this design is not representative of improved designs in general use, it was decided that testing of Type A meters should be discontinued. Subsequent tests were made on a meter design (Type B, Figure 7) that provides better thermal insulation for the slug and that more closely approximates the thermal characteristics of the test plates.

Figures 11 through 15 show the most significant results of the series of tests incorporating the Type B meters. Again, the reference temperatures used as a data control to show reproducibility between flux standardization tests and meter tests are tabulated in Appendix C, indicating an average spread in data of ± 2% from test to test.

Figures 11 and 12 compare the heat flux indicated by the Type B meter to the standard flux for heating by convection. Figures 13, 14, and 15 compare the heat flux indicated by the Type B meter to the standard flux for heating by radiation. It is apparent that the flux indicated by comparable meters varies in a different manner when heating is by radiation than when heating is by convection. In all cases, the flux indicated by the meter is initially about 20% higher than the true flux. All tests incorporating a meter of the same thickness as the test plate in which it is mounted show the error decreasing from this initial 20% as the temperature increases. When the heating mode is radiant, the meter reading approaches the true flux; when the heating mode is convective, the meter reading decreases until it is as much as 12% below the true flux.

The results shown for the case of a 1/4-inch meter in a 1/8-inch test plate (Figure 14) indicate the same initial error of about 20%; however, this error does not decrease as the temperature rises but remains almost constant.

Figure 15 shows the results of a test with radiant heating in which the true heating rate is about 17 rather than 22 Btu/ft^2-sec. This test indicates the same trends as those in Figure 13, which incorporated the same meters. However, the percentage errors are larger, the absolute magnitudes of the error being the same as for the higher flux.

DISCUSSION OF RESULTS

A. Type A Meter

The primary reason for the large errors in the Type A meters was conduction losses from the slugs to the stainless-steel ring, the ring being excessively massive. As a result, all of the meter readings are low since the insulation is inadequate to prevent such losses. It should be noted that the errors shown in Figure 10 for a 1/4-inch slug are much less than those shown in Figures 8 and 9 for 1/8-inch slugs. This is to be expected, since the 1/4-inch slug more closely approximates the thermal capacity of the steel ring and, as a result, the temperature difference across the alumina is much less. Hence, the heat losses are less, as are the errors.

With careful analysis of the data presented, other significant features can be seen. However, these same features are shown more clearly by analysis of the data taken with the Type B meter. The following discussion of the Type B meter applies, in general, to the Type A and therefore will not be repeated here.

B. Type B Meter

Radiant-heating tests for a 1/8-inch meter in a 1/8-inch plate, presented in Figure 13, show that the reading is initially high by about 16%. The error decreases with increased temperature until the meter agrees with the true flux at the conclusion of the test. Tests under the same conditions for a 1/4-inch meter in a 1/8-inch plate, presented in Figure 14, show that the meter reading varies from 23% to 26% high. Comparable tests with convective heating of very nearly the same rate are shown in Figures 11 and 12. The error in the 1/8-inch meter varies from +16% (the same as for radiation)to -12%, while the error in the 1/4-inch meter is from 19% to 22% high.

Comparison of the above radiant and convective tests indicates that the results are very similar but that the magnitudes of the errors are somewhat different. In both cases, flux indicated by the 1/8-inch meter decreases with increased temperature while that indicated by the 1/4-inch meter remains about constant.

One of the most pronounced effects shown in all tests is that the data from the meter and from the standardizing plate do not agree initially. It might be expected that the temperatures throughout the system would be near the same value at the start of heating and that the errors would consequently be zero or near zero. However, it was found

(Appendix A) that within the first 1/2 second of heating, sufficient temperature differences exist between the slug and stainless-steel plate and the surface of the insulation, due to the difference in thermal conductivities, that a large percentage of the heat absorbed by the insulation is conducted to the slug and steel plate rather than into the insulation. As heating continues, the relationship of the plate, insulation, and slug temperatures changes. The rate of heat conduction to the slug changes and the error in the meter reading varies. It is apparent from nearly all tests that the slug heats faster than its surroundings and that the rate of heat gain from the insulation decreases, with a resulting decrease in error. The only exception to this is the case of a 1/4-inch meter in a 1/8-inch plate. In this instance, the slug heats slowly and continues to absorb heat from the insulation.

In order to substantiate this theory, a meter was prepared that did not have its insulation blackened, the idea being that if less heat is absorbed by the insulation, less will be conducted to the slug, and the error should be less. The results of test 92, shown in Figure 13, bear out this hypothesis. The heat flux indicated by the meter was lowered by 8 to 10% except at the start of heating.

The errors caused by conduction from the insulation may prove to be a serious problem and one that cannot be overcome in meters incorporating an exposed insulation around the meter slug.

The fact that the meter output varies somewhat differently when heated by convection rather than radiation can be attributed to one or both of two effects. It may be that the temperature variation along the surface alters the heat transfer coefficient in the vicinity of the meter by disturbing the temperature gradients in the thermal boundary layer. It may also be that the difference is due to a change in conduction between the insulation and the slug caused by dissimilar heating of the insulation surface. As the insulation surface temperature rises, the temperature difference for convective heating is decreased, with a resulting decrease in heat absorbed by the insulation. With radiant heating, the heat absorbed by the insulation is constant. Under both conditions, the re-radiated energy is about the same. It can thus be seen that the insulation receives less net heat under convection than under radiation, which may well affect the lateral conduction between the slug and the insulation.

Whether the difference in results obtained under convection and radiation is primarily due to boundary-layer effects or to dissimilar heating effects cannot be determined from the data available. If the latter effect predominates, the results obtained may be peculiar to the meter design tested.

The volumetric heat capacity of stainless steel and copper are within about 5% of the same value.* Thus, a 1/8-inch meter in a 1/8-inch plate is a relatively go thermal match. Even with such a thermal match, however, the errors in the met readings are as large as 16% under both convection and radiation, as noted earlier

Figure 12 shows results of tests run using a 1/4-inch meter in both 1/8- and 1/4-inch plates with convective heating. The flux indicated in the two cases is entirely different and is a very clear demonstration that the accuracy of a meter this type is strongly dependent upon the geometry of the structure in which it is pl

The results shown in Figure 15 for a low radiant-heating rate, although not c plete enough to be conclusive, indicate that the accuracy of the meter tested may dependent on the magnitude of the heating rate. The initial error is about 28% in case, as opposed to about 15% for the same meter under a higher heating rate. I accuracy of this type meter is, in fact, dependent upon heating rate, such variati heating rate must be considered in carrying out meter calibration.

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* ρC_p for copper = 51.3 Btu/ft^3-°F. ρC_p for stainless steel = 54.3 Btu/ft^3-°

CONCLUSIONS

1. The trends of the errors in the heat flux indicated by the slug-type meters tested (Type B) are similar under radiation and convection but the magnitudes are somewhat different. This can be attributed either to convective boundary-layer effects or to dissimilar heating of the insulating ring, or both, the latter effect being design dependent.

2. Even though the thermal capacities of the slug and of the structure in which it is mounted may be similar, the presence of the insulating support material for the slug may give rise to sufficient perturbations to cause significant cross-conduction effects within the meter assembly and hence an error in indicated flux. Such conduction effects may be caused by rapid heating of the exposed insulation surface surrounding the slug and separating it from the structure in which it is supported.

3. The accuracy of the Type B meter tested appears to be related to the magnitude of the flux; i.e., the lower the flux the greater the percentage of error.

RECOMMENDATIONS

1. Typical meters currently used in missile tests should be tested under both convective and radiant heating to determine whether or not the convective effects that have been observed can be expected in all designs.

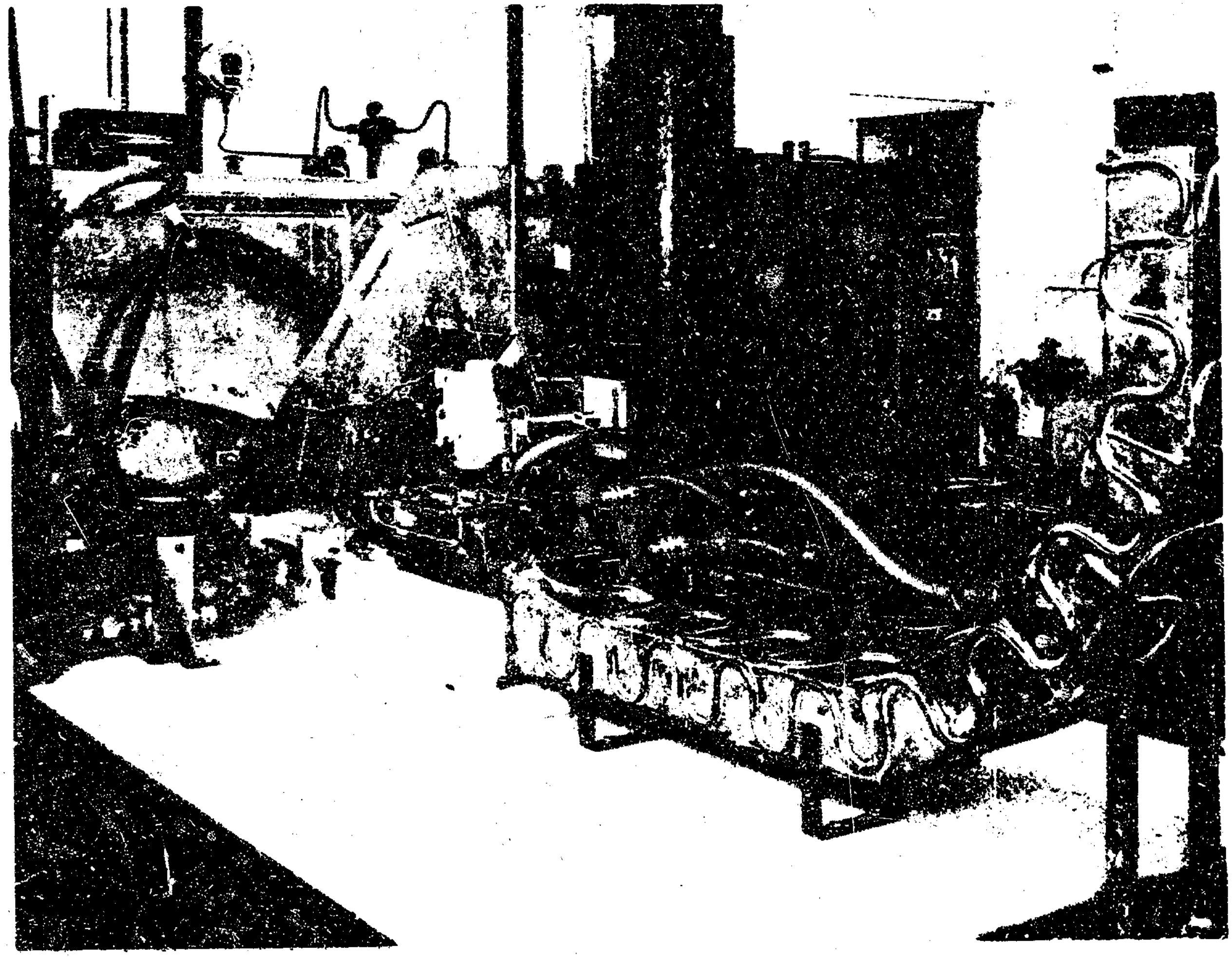
2. These same meters should be tested under both radiation and convection heating in non-metallic structures where larger surface-temperature discontinuities will be encountered. Since these meters are commonly used in such installations, the convective boundary-layer effect should be investigated.

3. Further studies should be made to determine the dependence of meter accuracy on heating rate.

4. The method that was used in making standard heat-flux measurements should be improved by using a thinner plate, perhaps as small as 1/32 inch.

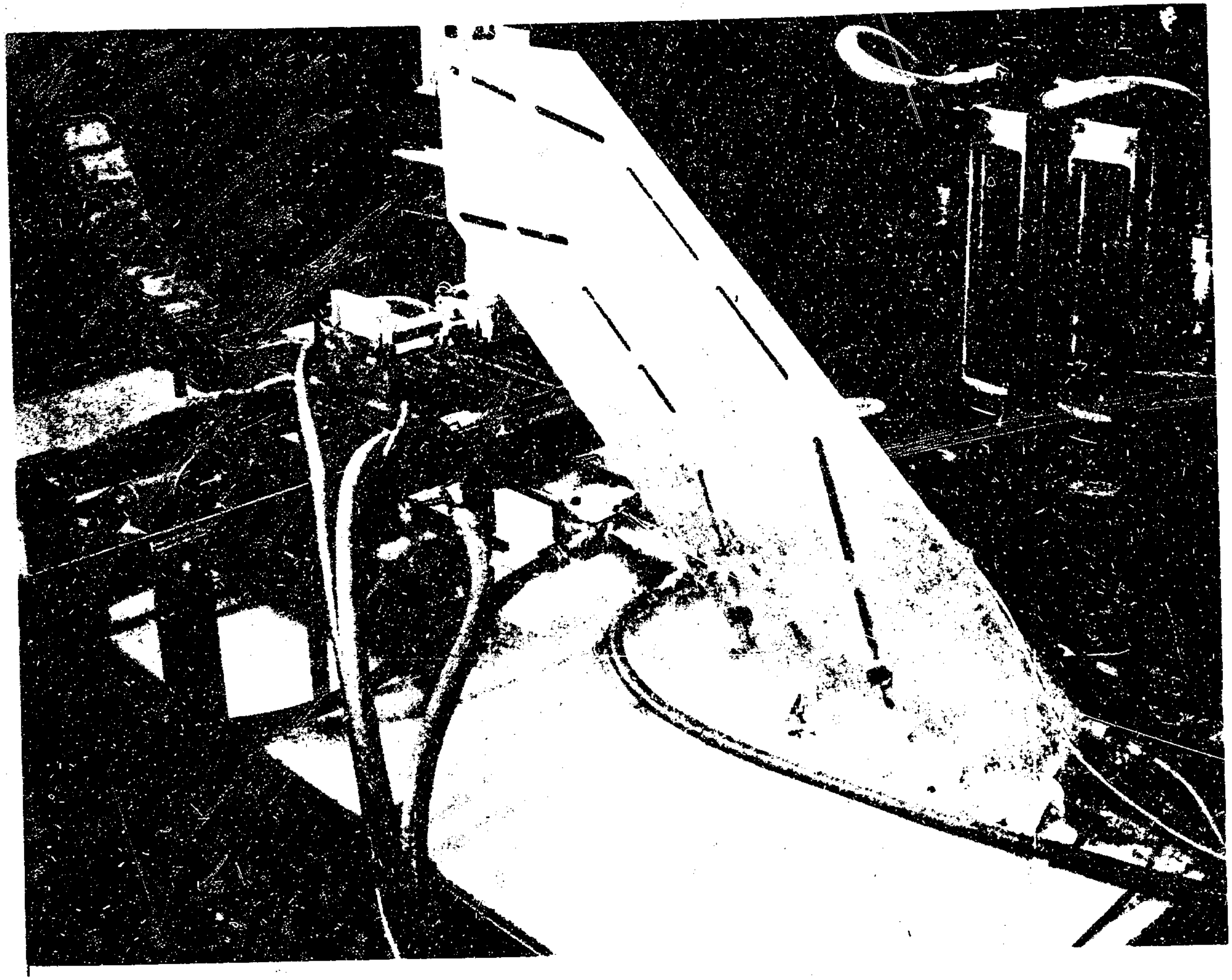
5. Studies should be made to provide the most appropriate and practical meter designs.

ILLUSTRATIONS



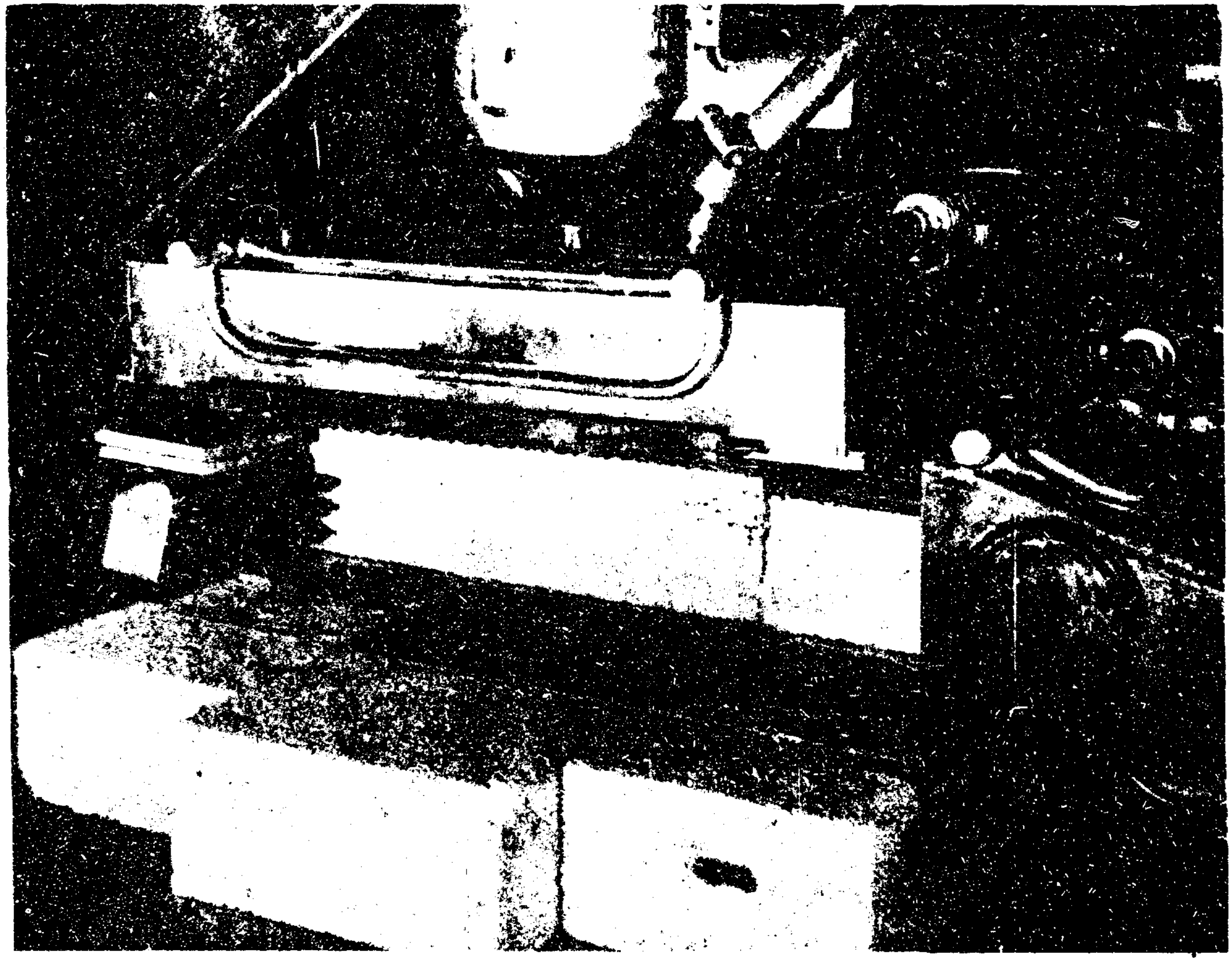
LABORATORY HEATING APPARATUS
FRONT VIEW

FIGURE 1



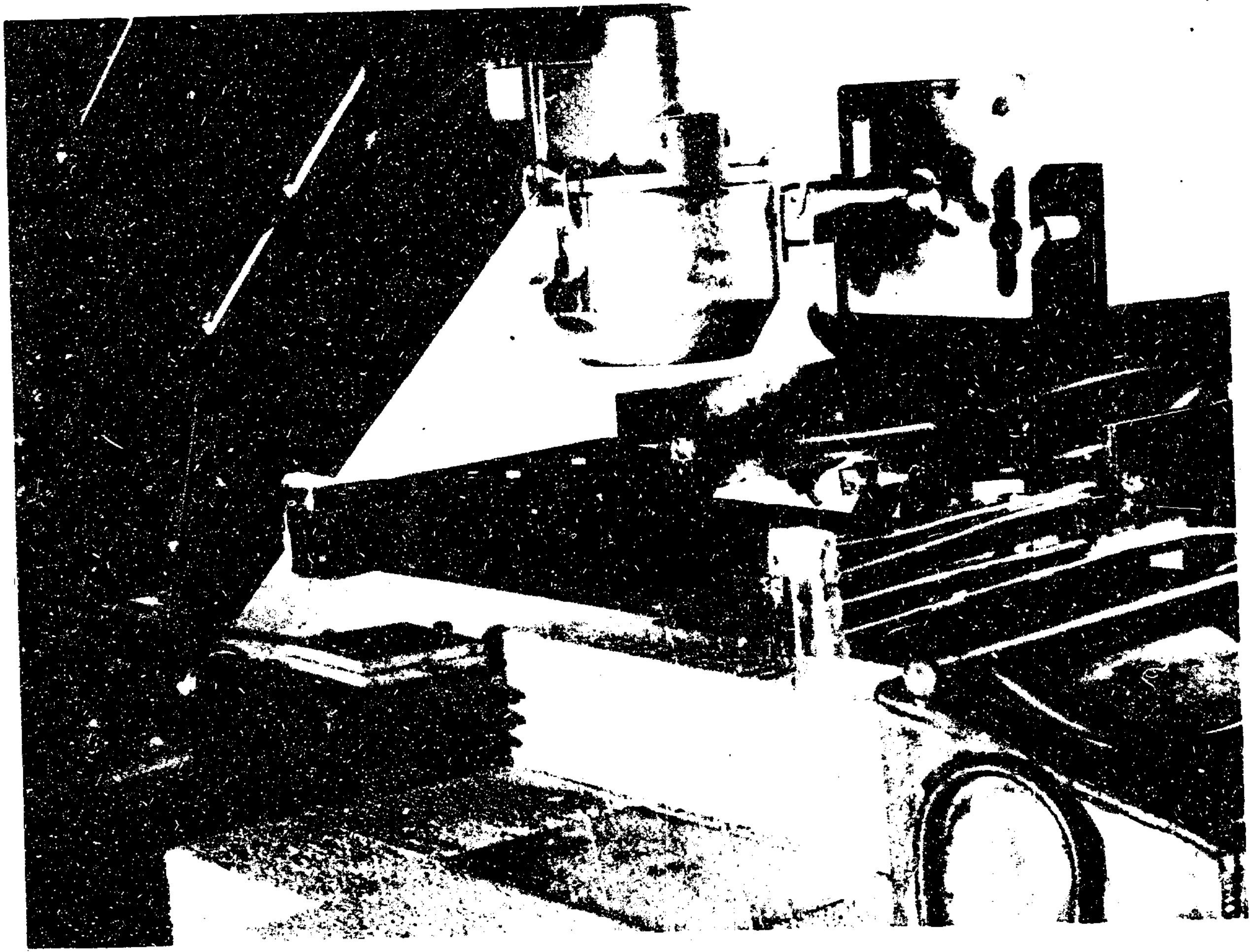
LABORATORY HEATING APPARATUS
REAR VIEW

FIGURE 2



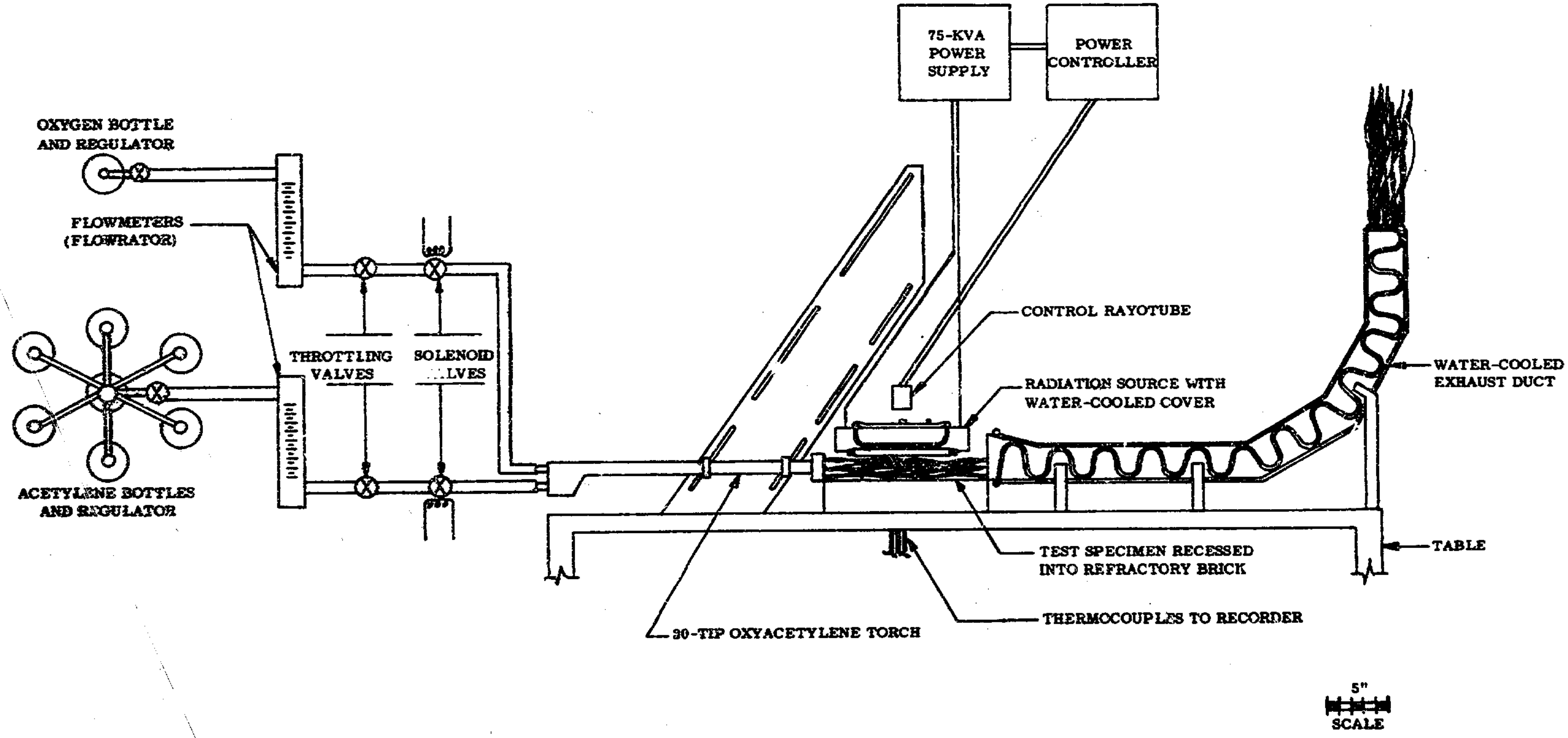
HEATING APPARATUS TEST SECTION

FIGURE 3



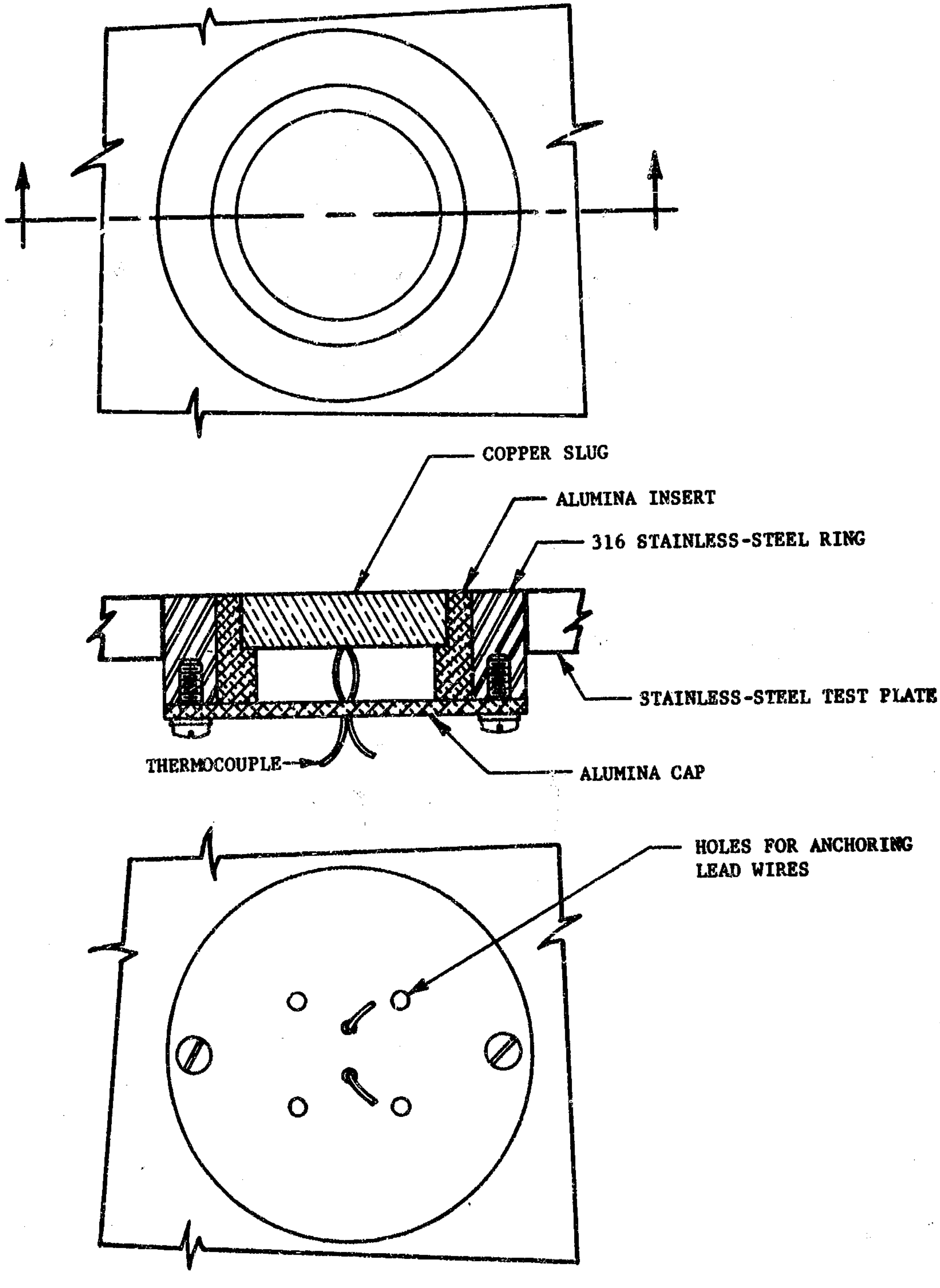
INTERNAL VIEW OF RADIANT HEATER AND TEST SECTION

FIGURE 4



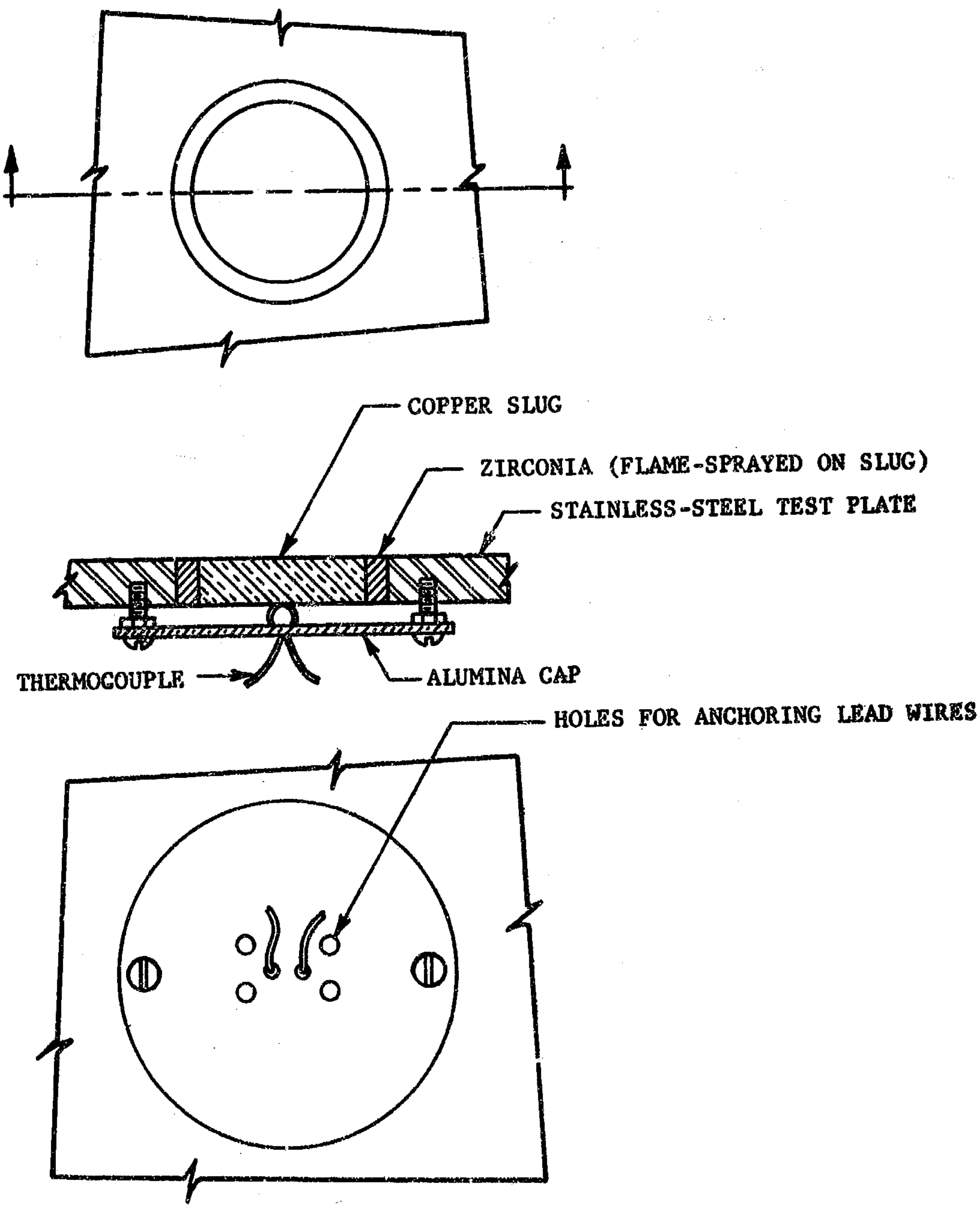
EXPERIMENTAL HEATING APPARATUS

FIGURE 5



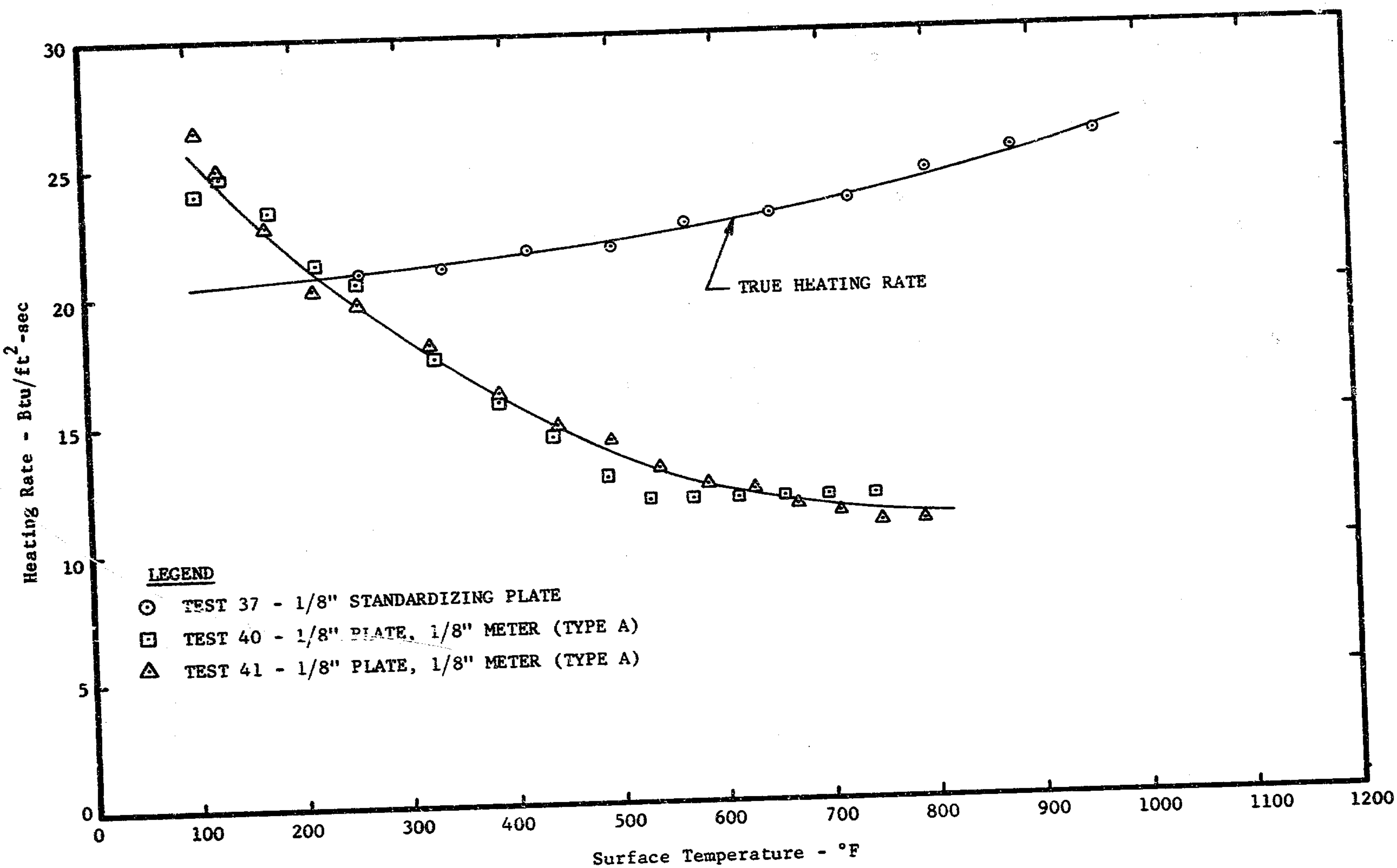
TYPE A SLUG-TYPE HEAT-FLUX METER

FIGURE 6



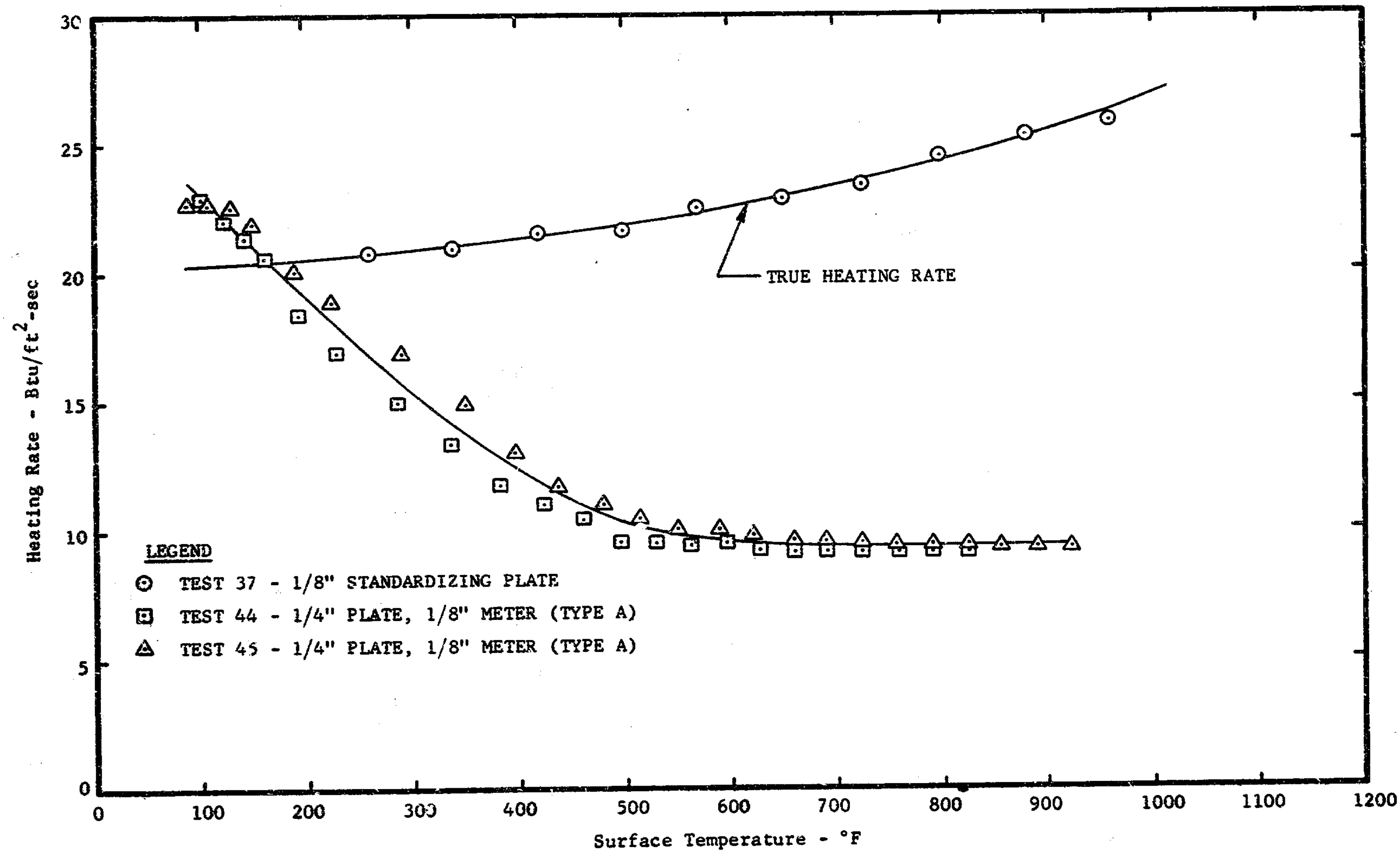
TYPE B SLUG-TYPE HEAT-FLUX METER

FIGURE 7



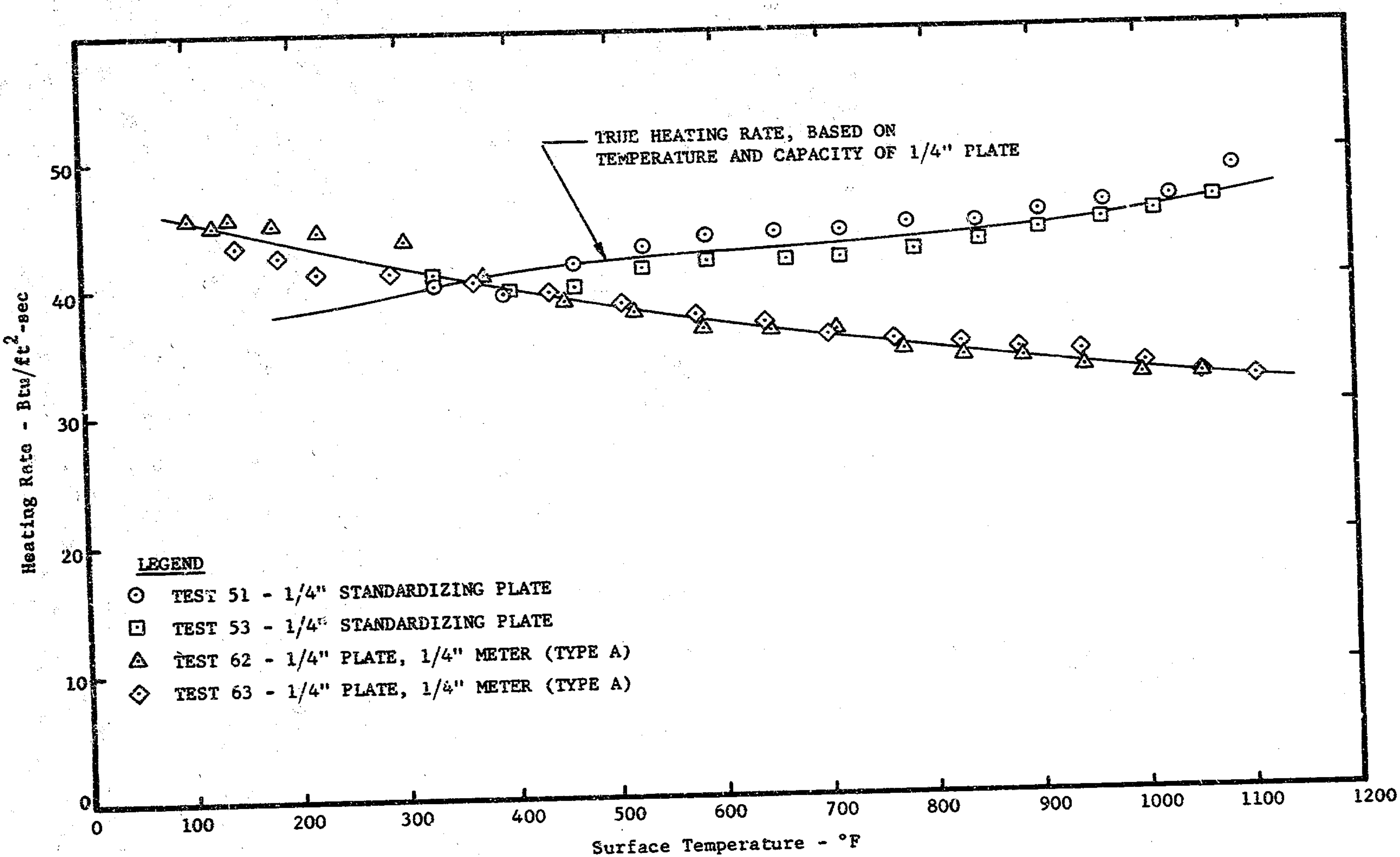
CONVECTIVE-HEATING-RATE MEASUREMENTS
(TYPE A METER)

FIGURE 8



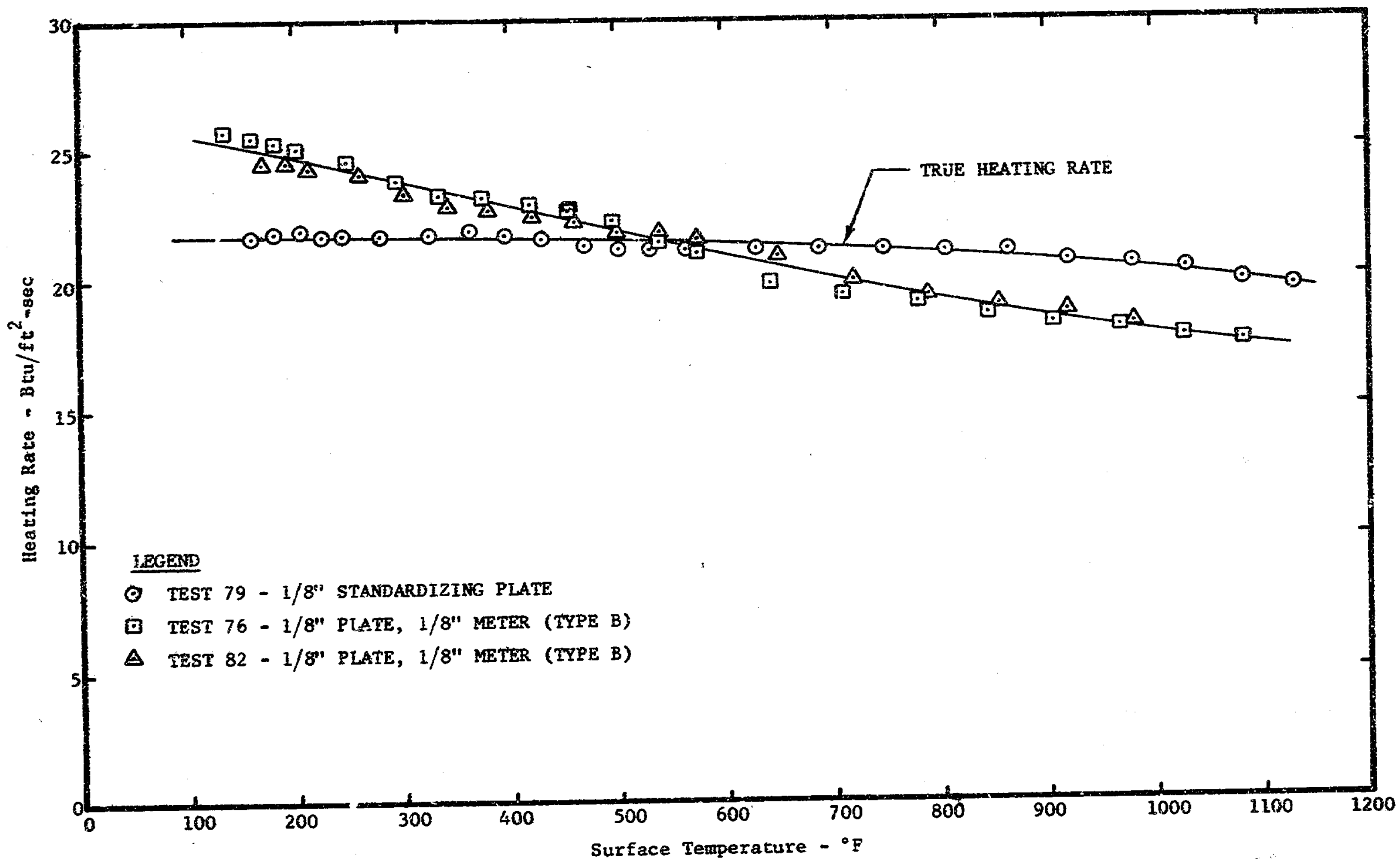
CONVECTIVE-HEATING-RATE MEASUREMENTS

(TYPE A METER)



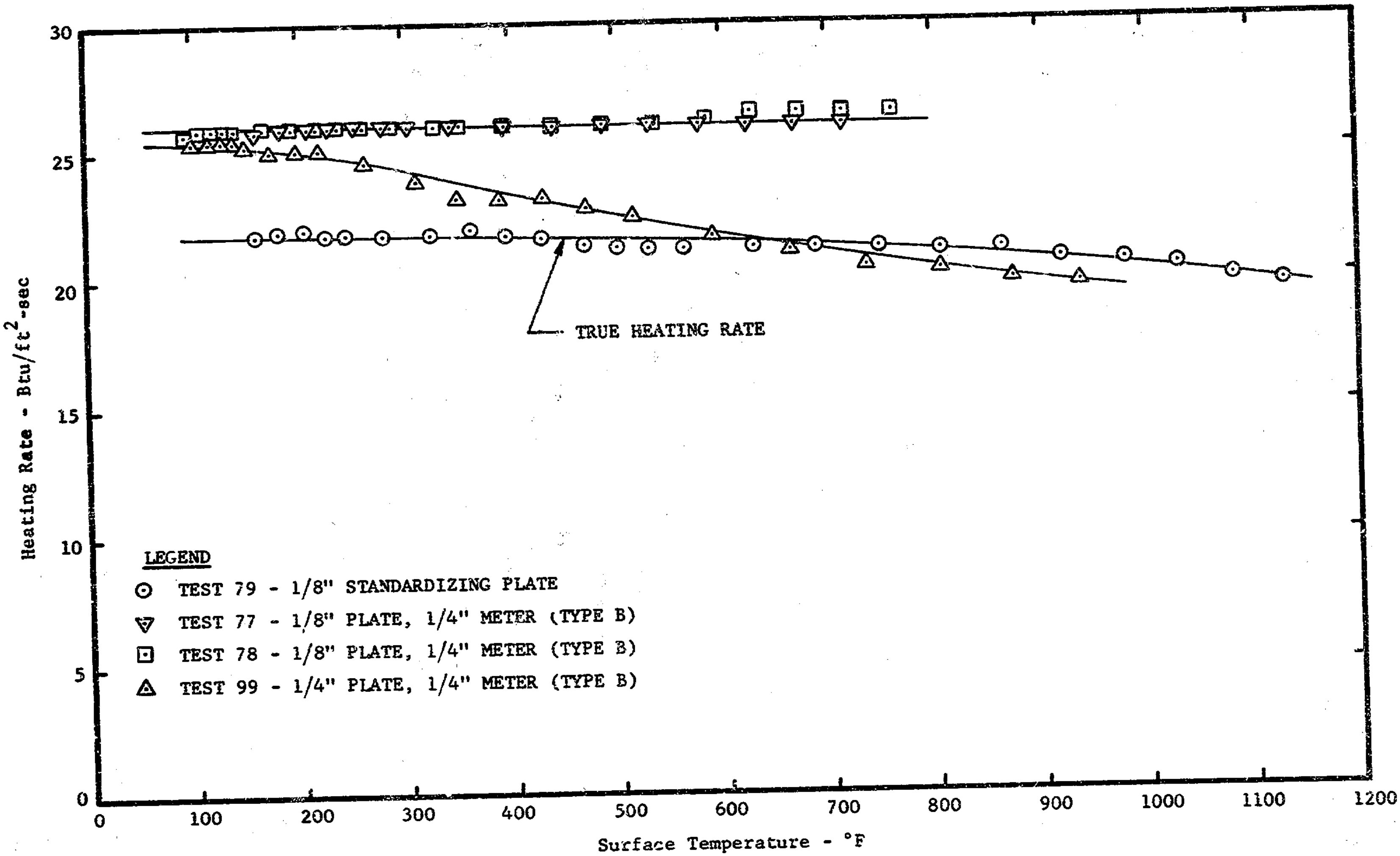
COMBINED CONVECTIVE-RADIANT HEATING-RATE MEASUREMENTS
(TYPE A METER)

FIGURE 10



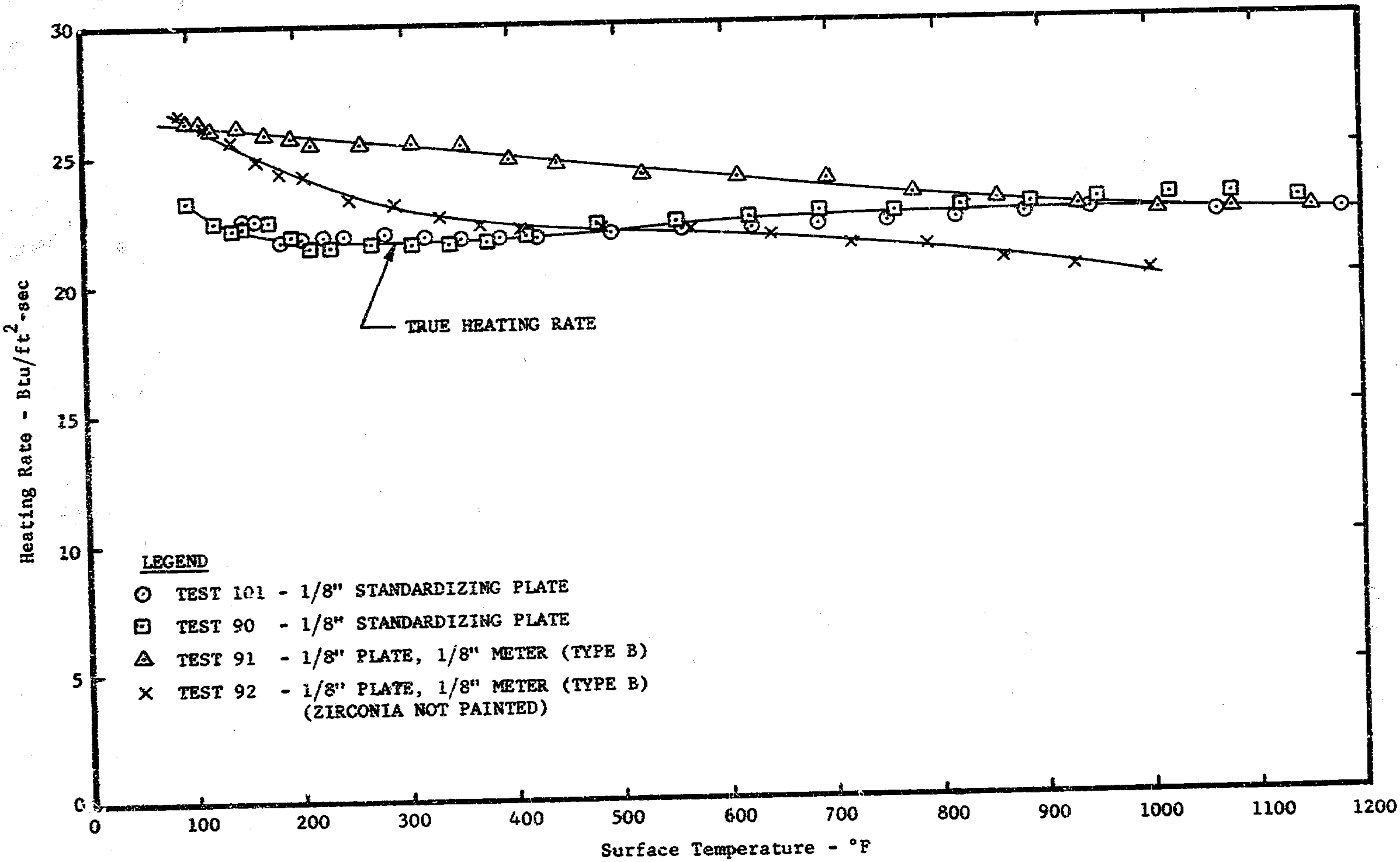
CONVECTIVE-HEATING-RATE MEASUREMENTS
(TYPE B METER)

FIGURE 11



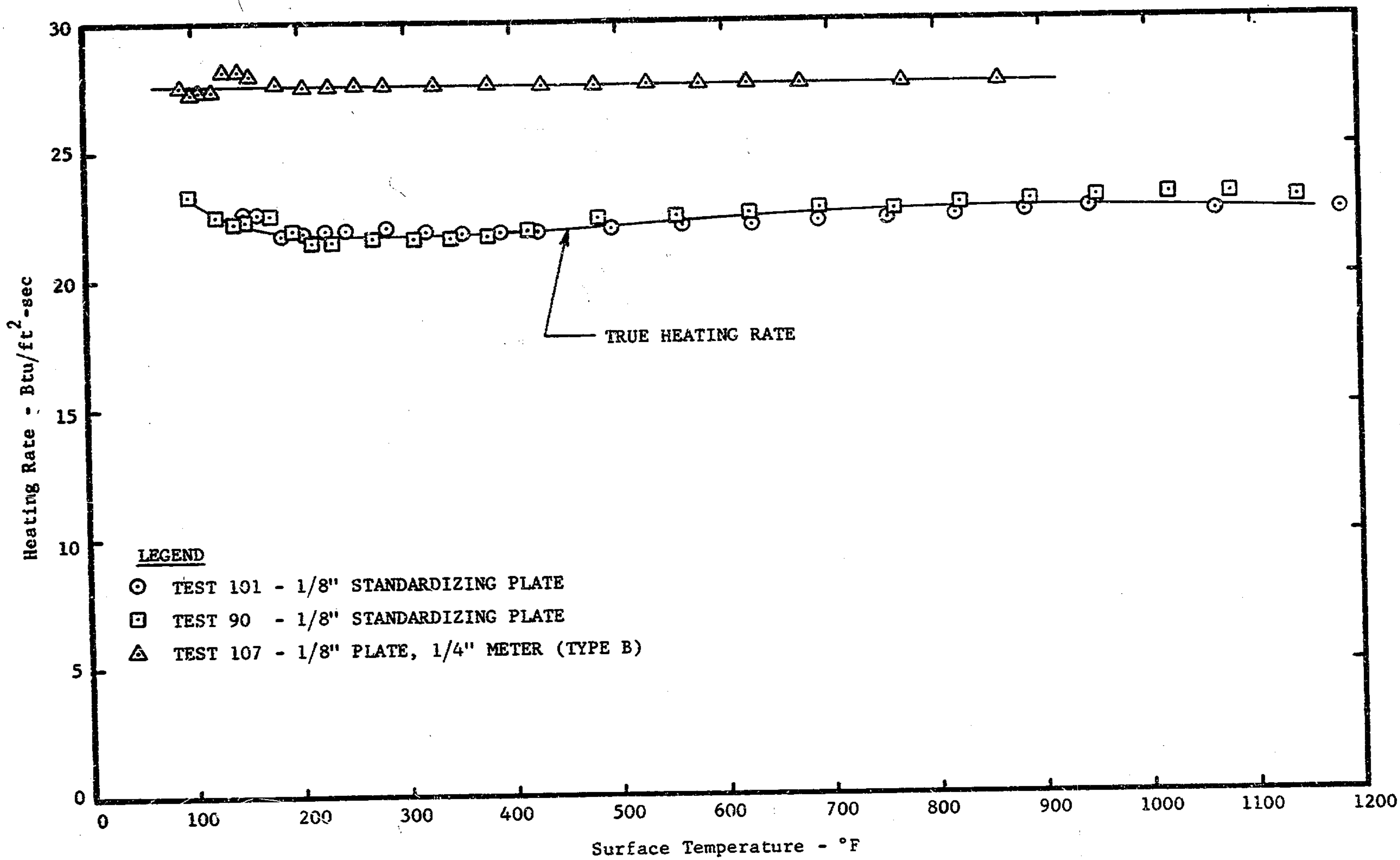
CONVECTIVE-HEATING-RATE MEASUREMENTS
(TYPE B METER)

FIGURE 12



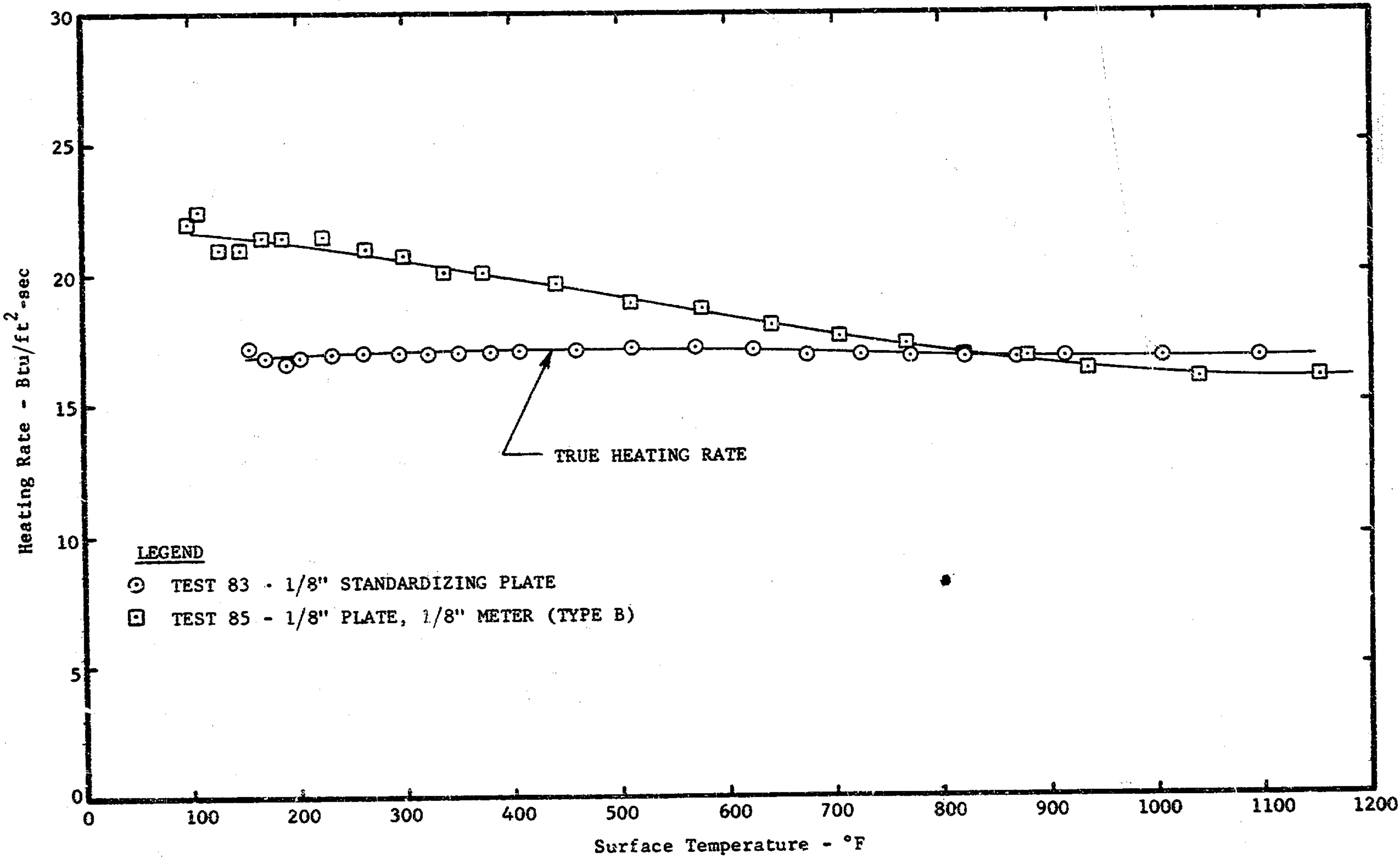
RADIANT-HEATING-RATE MEASUREMENTS
(TYPE B METER)

FIGURE 13



RADIANT-HEATING-RATE MEASUREMENTS
(TYPE B METER)

FIGURE 14



RADIANT-HEATING-RATE MEASUREMENT
(TYPE B METER)

FIGURE 15

APPENDIX A

FURTHER ANALYSIS OF METHODS AND RESULTS

1. Lateral Conduction in the Standardizing Plate

In measuring the true heating rate with a thin steel plate, it was assumed that the heating rate was equal to the sum of the heat stored and the heat lost at the rear surface, and that lateral conduction within the plate was insignificant. This can be shown to be a valid assumption by the following discussion.

The reference temperature was measured 1 inch from the center of the plate. The difference between the temperature at this point and the surface temperature at the center is indicative of the gradients that exist. In the case of convective heating, the reference temperature is higher than the surface temperature at the center; thus heat is conducted to the center while at the same time it is conducted away in the downstream direction. Since heating is quite uniform over the center 2 inches of the plate, conduction in the direction normal to flow should not be large.

With radiant heating, on the other hand, the center of the plate receives more heat than any other point (see following section) and heat is conducted away equally in all directions. Thus, the errors due to lateral conduction should be much less under convective heating than under radiant heating. The measured temperature difference between the center and reference points on the radiation tests can be seen to increase approximately linearly over the test period. Initially, there are no lateral gradients, and consequently no errors result. As the test proceeds, the gradients and resulting errors build up and are greatest at the end of the test.

Midway through the tests, it is seen that the measured temperature difference under radiation is about 30°F. This difference exists over a 1-inch distance, and it can be assumed that a local gradient of 30°F per inch exists at the center of this 1-inch span or around the circumference of a 1/2-inch radius disc. Based on this assumption the ratio of the rate of heat conducted out of this disc to the heat input can be computed.

The heat conducted out is given by

$$\dot{Q}_o = 2\pi R \ell k \left.\frac{dT}{dr}\right|_R ,$$

where R = radius of disc considered (1/2 inch),

ℓ = thickness of disc considered (1/8 inch),

k = thermal conductivity,

T = temperature,

r = polar coordinate;

and the heat input is

$$\dot{Q}_i = \dot{Q}\pi R^2 \quad ,$$

where $\dot{Q}$ = imposed heating rate.

Taking the heating rate to be 22 Btu/ft^2-sec, the desired ratio is

$$\frac{\dot{Q}_o}{\dot{Q}_i} = \frac{2\ell k \left.\frac{dT}{dr}\right|_R}{\dot{Q} R} = 0.023 \quad .$$

The lateral conduction can therefore account for as much as a 2.3% error in the assumed true flux midway through the test. This error is smaller earlier in the test and may approach 5% at test conclusion. The resulting errors will be much less with convective heating.

2. Shape-Factor Distribution over Test Plate

The test plates were centered directly under the graphite heater element. The view angle from the plate to the heater was therefore largest at the center and decreased slightly at points away from center. The resulting uneven heating was primarily the cause of the lateral conduction errors shown above. The shape factor at the center of the plate, the plate being 4.0 inches from the heater, was 0.416,[4] while at a point 1 inch away from the center along either centerline, it was 0.398. The difference is about 4%, which results in 4% less heat flowing to the plate 1 inch from the center. This shape-factor distribution is therefore the primary cause of the lateral conduction losses.

3. Rear-Surface Heat Losses from Standard Plate

In making the standard flux measurements with a 1/8-inch-thick plate, the heat loss from the rear surface of the plate was considered. The back of the plate was

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4. W. H. McAdams, Heat Transmission, New York, McGraw-Hill Book Co. (1954).

packed with Fiberfrax,* a high-temperature low-density insulation. The insulation was approximately 3 inches thick and thus could be considered semi-infinite. Since the temperature of the rear surface of the plate could be considered to be the surface temperature of the semi-infinite insulation, the conduction loss could be computed using techniques reported in the Phase 1 final report.[1] It was found, however, that this method was too laborious to use on numerous tests; a simpler approximation was therefore sought.

It was observed that the measured temperature rise at the plate/insulation interface was very nearly linear with time. The surface temperature of the insulation could therefore be assumed linear and the loss calculation greatly simplified.

The temperature distribution within a semi-infinite solid whose surface temperature is a linear function of time (At) is

$$T = 4\, At\, i^2 \operatorname{erfc} \frac{x}{2\sqrt{\alpha t}} \quad ,^5$$

where T = temperature, t = time, x = distance measured from the surface, and α = thermal diffusivity. The temperature gradient at the surface (x = o) is

$$\left.\frac{dT}{dx}\right|_{x=o} = -1.128\, A \left(\frac{t}{\alpha}\right)^{\frac{1}{2}} \quad .$$

The heat flux passing x = o is then

$$\dot{Q}_\ell = -k \left.\frac{dT}{dx}\right|_{x=o} \quad \text{or} \quad \dot{Q}_\ell = 1.128\, Ak \left(\frac{t}{\alpha}\right)^{\frac{1}{2}} \quad ,$$

where k = thermal conductivity. This operation is simple, being a constant times $t^{\frac{1}{2}}$, and yields values well within the accuracy of other measurements.

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* A product of the Carborundum Company.

5. H. S. Carslaw and J. C. Jaeger, Conduction of Heat in Solids, 2nd Edition, Oxford Press, (1959).

4. Variation of Heat Flux with Time

The true-heating-rate curves shown in Figures 8, 9, and 10 are seen to increase appreciably during the course of the tests. It was found that a refractory brick that was placed upstream of the test plate and over which the flame passed prior to reaching the plate became extremely hot by the end of the test. The result was less cooling of the boundary-layer gases as the test progressed, so that more heat was available to heat the plate.

This situation was corrected on all Type B meter tests by replacing the brick with a stainless-steel plate of the same thickness as the test plate. It is important that the plates be the same thickness so that there are no discontinuities in the temperature of the surface over which the gases pass prior to reaching the instrumented portion of the plate. The addition of the upstream plate was a satisfactory solution to the problem.

5. Conduction between Slug and Insulation

One of the major points brought out by the results of the test program was the existence of gross errors immediately after initiation of heating. As previously noted, this error has been attributed to conduction from the insulation ring to the slug.

Because of the low thermal conductivity of the insulation (zirconia in the Type B meter), its surface temperature rises very rapidly since the absorbed heat cannot be conducted away from the surface. At the same time, the surface temperature of the copper slug rises slowly, since all absorbed heat is diffused rapidly throughout the slug. As a result, a large temperature gradient exists between the surface of the insulation and the adjacent copper slug. Since all heat absorbed by the insulation must be either stored or conducted away, it is conducted primarily to the copper along the path of least resistance. Some is, of course, conducted to the steel plate, but since the conductivity of this plate is also low compared with that of copper, the copper absorbs a much greater proportion.

The large gradients that can potentially exist between the zirconia surface and the copper and steel can be shown by computing the temperatures that would exist in each when heated at the same rate for a period of 1 second, assuming that no lateral conduction occurs during this period. Considering the 1/8-inch-thick meter in a 1/8-inch-thick plate, each material can be treated as a 1/8-inch-thick flat plate heated at a constant rate (20 Btu/ft^2-sec) on one surface and insulated on the other. The sol for the temperature distribution is well known[5] and is as follows:

$$T - T_o = \frac{qt}{\rho C_p \ell} + \frac{q\ell}{k}\left[\frac{3x^2 - \ell^2}{6\ell^2} - \frac{2}{\pi^2}\sum_{n=1}^{\infty}\frac{(-1)^n}{n^2} e^{-\alpha n^2 \pi^2 t/\ell^2} \cos \frac{n}{}\right.$$

The assumed properties are listed below.

Material	k (Btu/hr-ft-°F)	ρ (lb/ft^3)	C_p (Btu/lb-°F)
Copper	223	558	0.092
Stainless steel	8	494	0.11
Zirconia	0.6	374	0.11

The temperature rise of each material was computed at the surface, midpla and rear after 1 second of heating. These data are tabulated below.

Location / Material	Surface Temp. (°F)	Midplane Temp. (°F)	Rear Temp. (°F)
Copper	38	36	36
Stainless steel	66	30	20
Zirconia	274	12	0

The above table shows clearly that extremely large gradients exist between copper and zirconia and also between steel and zirconia. Although the area over which these gradients exist is small, the gradients are steep. The temperature difference between zirconia and copper may, in actuality, be about 200°F, and th distance over which this difference exists would be no more than 1/32 inch since the zirconia is only 1/16 inch thick. This temperature difference suggests the p bility of a gradient of the order of 6400°F per inch. Even though the area of con is small, such an extreme gradient will provide a large amount of conduction he

As time passes, the temperatures should tend to equalize, since the copper then heat the zirconia near the unheated rear surface. This fact and the fact tha the steel will begin to absorb a larger proportion of the heat as the copper becon hotter explain why the errors shown by the meter decrease with time.

Since the complete analytical evaluation of the conduction phenomena is rather complex, it can be carried out practically only by means of a computer solution, which was beyond the scope of this program.

6. Other Possible Sources of Error in Meters

In addition to the previously described sources of error in the heat flux indicated by the meters tested, there are other possible sources of smaller errors. The indication that an unexpected effect may be altering the meter readings is shown in Figures A-1 and A-2. The plate shown was heated convectively twice prior to the time the photograph of Figure A-2 was taken. Gas flow was from left to right in the photograph. The paint is seen to be badly worn on the meter and for a distance slightly more than an inch downstream of the meter. This is evidence that the meter has had some perturbing effect on the flow. Whether the effect is hydrodynamic or thermal cannot be concluded from the photograph; however, it appears more likely to be the former. If this is so, it is indicative of the care that must be taken in a meter installation. The meter pictured was installed with smoothness in mind and the resulting roughness was no more than a few thousandths of an inch. Apparently turbulence was still induced.

Another source of error that can be expected in almost any meter is rear-surface heat losses from the slug. Near the conclusion of the tests run, the slugs were at temperatures of the order of 1000°F. Assuming the emissivity of the rear of the slug (oxidized copper) to be about 0.5,[4] the radiant loss is about 1 Btu/ft^2-sec or about 5%. The effect of convective losses is probably small, since there is essentially a dead air space behind the slug.

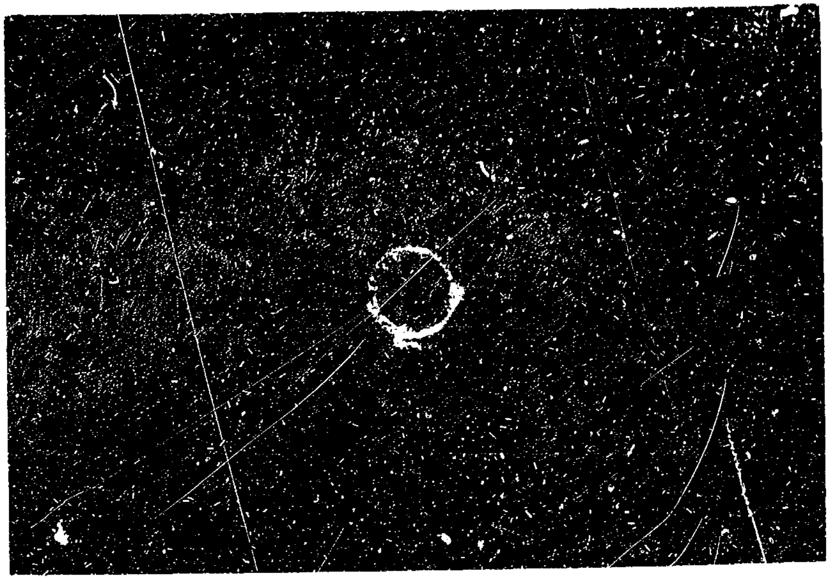
7. Reference Temperatures

The reference temperatures used as data control were, in general, reproducible. Occasional trouble arose due to such things as excessive oxidation of the graphite heater, which showed up immediately in the reference temperature. In such instances the test was invalidated.

A typical plot of the reference temperatures used to correlate a set of tests is shown in Figure A-3. These curves were used in establishing the validity of the comparisons made in Figures 11 and 12, except for test 99, in which test the plate was a different thickness.

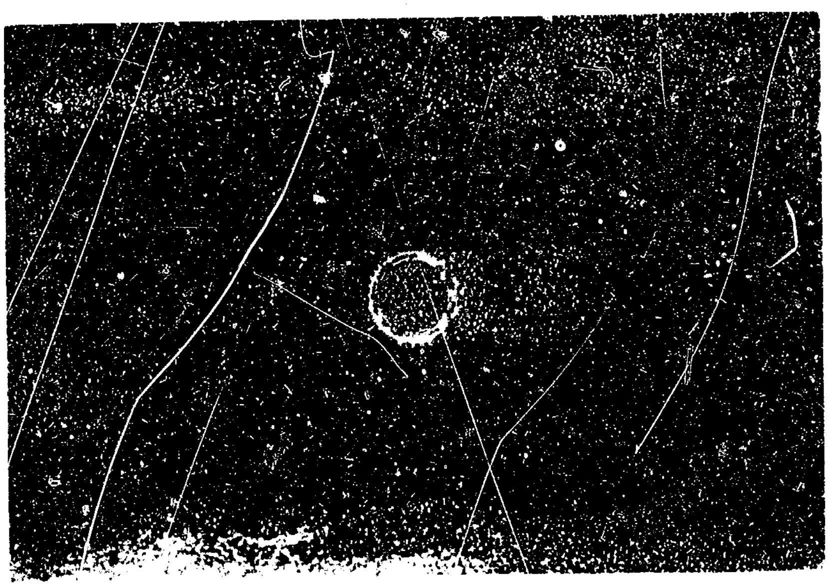
In switching from 1/8- to 1/4-inch-thick plates, it was also necessary to assure reproducibility. The best assurance was if tests prior to and after the switch gave consistent results. As an added check, a 1/4-inch plate, with a reference thermocouple and surface and rear thermocouples at the center of the plate, was heated. The heating rate to the plate was computed in the same manner as for the 1/8-inch standard plate. If the computed heating rate compared favorably with the previous 1/8-inch-plate measurement, it was assumed that heating conditions had been reproduced; the reference temperature was then taken as the standard for subsequent 1/4-inch-plate tests.

It should be noted that the temperatures measured in test 37 were both rear-surface measurements, since construction of the thermocouple plugs had not been completed. However, the validity and consistency of the results shown can be justified by comparison of the tabulated temperatures from other tests.



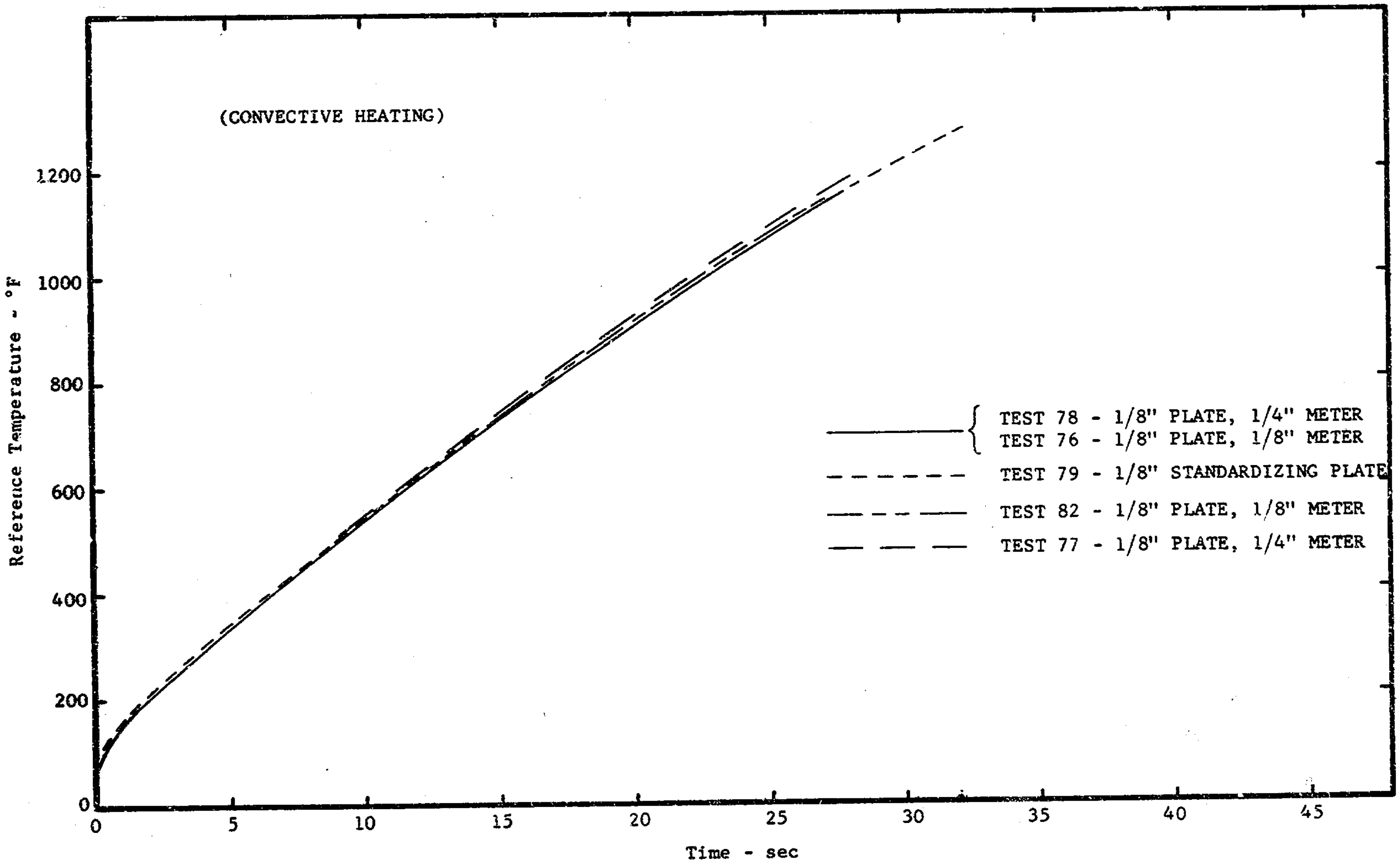
TEST PLATE WITH METER INSTALLED
BEFORE HEATING

FIGURE A-1



TEST PLATE WITH METER INSTALLED
AFTER HEATING

FIGURE A-2



TYPICAL REFERENCE TEMPERATURE COMPARISON

FIGURE A-3

HIGH TEMPERATURE HEAT CONTENT OF 316 STAINLESS STEEL

Final Report

June 4, 1962

TO: Advanced Technology Laboratories, Sponsor

FROM: Institute of Engineering Research
University of California, Berkeley

PROJECT: Service to Industry Contract UCB-Eng-6560

PERIOD OF PERFORMANCE: April 1, 1962, to May 31, 1962

FACULTY INVESTIGATOR: Ralph Hultgren, Professor of Metallurgy

PERSONNEL: Raymond L. Orr, Assistant Research Engineer
Weston B. Kendall, Graduate Research Engineer
Robert Joseph, Engineering Aide

Abstract

High temperature heat contents with respect to 25°C were determined for two samples of 316 Stainless Steel between 108° and 1083°C. The results are completely regular and are in very good agreement with previously available data for 316 Stainless Steel. Tables of the experimental data and smoothed values are presented together with a brief discussion of the method and results.

Experimental

Spherical specimens weighing about 0.8 gm. were taken from samples of 316 Stainless Steel rod stock and plate stock supplied by the sponsor. Each specimen was enclosed in a platinum foil capsule weighing about 0.6 gm.

High temperature heat contents were measured in an isothermal Bunsen-type calorimeter using diphenyl ether as the working substance. A complete description of the apparatus and techniques has been given previously[1] and will not be repeated here. Runs were made at intervals of approximately 100°C between 108° and 1083°C. Duplicate measurements were made on the "rod" specimen at most temperatures. Measurements on the "plate" specimen were made at four temperatures spaced fairly evenly over the entire range of measurement.

Results

The experimental data for the rod and plate specimens are listed in Table I. In order to correct the results for the difference in temperature between the melting point of diphenyl ether, 26.9°C, and the standard reference temperature, 25°C, the room temperature heat capacity extrapolated from the data, 0.108 cal/°C-gram, was used.

Data from the rod and plate specimens are in agreement well within the overall scatter of the results. A plot of all the data follows a smooth curve within an average deviation of about 0.4%. Smoothed values taken from the selected curve are given in Table II. The listed values should represent the heat contents of either the rod or plate samples within a maximum uncertainty of ± 0.5%.

The data are in quite good agreement with the previous results for 316 Stainless Steel obtained by Fieldhouse, Hedge, and Lang.[2]

References

(1) Ralph Hultgren, Peter Newcomb, Raymond L. Orr, and Linda Warner, Proceedings, Symposium No. 9, National Physical Laboratory: The Physical Chemistry of Metallic Solutions and Intermetallic Compounds, H.M.S.O., London, Paper 1H.

(2) Fieldhouse, I.B., J.C. Hedge, and J.I. Lang, WADC Tech. Rept. 58-24, November 1958.

TABLE I

Experimental Data - Heat Content Measurements on 316 Stainless Steel Rod and Plate Samples

(Note: Runs marked (P) are for plate sample, remainder are for rod sample.)

Temp., °C	$h_T - h_{25°C}$ cal./gram	Temp., °C	$h_T - h_{25°C}$ cal./gram
108.1	9.07	724.2	90.74
204.1	20.23	(P) 825.8	105.57
(P) 204.6	20.96	825.8	105.33
204.7	20.74	827.4	105.11
299.0	31.94	844.6	108.24
299.2	32.33	848.4	108.90
395.2	44.86	988.4	129.34
395.4	44.73	988.4	130.22
(P) 506.3	59.87	1040.8	139.13
506.4	60.20	1082.5	142.95
506.5	59.96	(P) 1082.6	144.89
593.3	79.09	1082.8	145.33
593.9	72.46		

TABLE II

Smoothed Heat Content Values for 316 Stainless Steel

Temp., °C	$h_T - h_{25°C}$ cal./gram	Temp., °C	$h_T - h_{25°C}$ cal./gram
100	8.31	700	87.14
200	20.01	800	101.66
300	32.43	900	116.54
400	45.50	1000	131.76
500	59.05	1100	147.39
600	72.96		

Temp., °F	$h_T - h_{77°F}$ BTU/lb.	Temp., °F	$h_T - h_{77°F}$ BTU/lb.
100	2.50	1200	143.73
200	13.60	1300	157.99
300	25.10	1400	172.44
400	36.99	1500	187.13
500	49.28	1600	201.97
600	61.96	1700	217.01
700	75.00	1800	232.26
800	88.34	1900	247.73
900	101.91	2000	263.43
1000	115.68	2100	279.36
1100	129.61		

APPENDIX C

TABULATED DATA

TABLE I

TEST 37

Plate thickness 1/8"

Time (sec)	Reference* Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	73	74			
$\frac{1}{2}$	75	76			
1	85	88			
2	114	118			
3	147	152			
4	182	187			20.8
6	256	258			21.0
8	332	328			21.6
10	406	396			21.7
12	482	464			22.6
14	557	535			22.9
16	628	606			23.4
18	702	675			24.5
20	778	744			25.3
22	851	815			25.9

Loss included with $q_{\ell} = 0.156\ t^{\frac{1}{2}}$

* Measured at back surface.

TABLE II

TEST 40

Plate thic
Slug thick

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	84	87	
½	141	107	24.0
1	174	132	24.7
2	227	176	23.3
3	275	218	21.2
4	322	258	20.6
6	418	329	17.5
8	502	388	15.8
1[illegible]	588	439	14.4
12	670	488	12.8
14	748	531	11.9
16	830	573	"
18	905	615	"
20	981	657	"
22	1055	698	"
24	1126	745	"

TABLE III

TEST 41

Plate thickness 1/8"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	85	82	
½	141	107	26.5
1	170	127	25.0
2	218	172	22.7
3	266	215	21.2
4	315	256	20.7
6	402	325	18.1
8	485	388	16.1
10	566	443	14.9
12	650	495	14.3
14	730	540	13.1
16	809	586	12.5
18	887	628	12.2
20	962	670	11.6
22	1032	710	11.3
24	1097	749	10.9
26	1165	789	10.9

TABLE IV

TEST 44

Plate thickness 1/4"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	77	77	
½	138	100	22.9
1	167	122	22.0
1½	188	140	21.3
2	207	159	20.6
3	236	192	18.4
4	265	227	16.9
6	318	285	15.0
8	365	335	13.4
10	407	381	11.8
12	445	423	11.1
14	488	459	10.5
16	528	495	9.6
18	567	528	9.6
20	607	562	9.5
22	646	595	9.6
24	684	626	9.3
26	723	659	9.2
28	764	690	"
30	799	724	"
32	839	757	"
34	875	789	"
36	915	824	"

TABLE V

TEST 45

Plate thickness 1/4"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	68	64	
½	128	84	22.7
1	156	106	22.7
1½	177	128	22.6
2	198	149	21.9
3	231	188	20.1
4	258	225	18.9
6	308	289	16.9
8	355	348	14.9
10	402	396	13.1
12	447	437	11.8
14	490	479	11.1
16	576	514	10.5
18	580	550	10.2
20	625	587	10.1
22	669	622	9.9
24	710	658	9.7
26	752	688	9.7
28	795	723	9.6
30	839	755	9.5
32	880	789	9.4

TABLE VI

TEST 51

Plate thickness 1/4"

Time (sec)	Reference Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	67	68		68	
$\frac{1}{2}$	148	68		129	
1	207	68		180	
2	278	79		247	
3	329	103		293	
4	375	132		334	40.0
6	451	199		400	39.4
8	519	269		466	41.7
10	587	340		532	42.8
12	649	411		592	43.2
14	715	480		655	43.8
16	778	549		717	43.9
18	844	615		780	44.5
20	909	677		846	44.4
22	970	742		906	45.2
24	1033	808		967	45.9
26	1094	872		1030	46.3
28	1154	938		1090	48.6

Loss included with $q_\ell = 0.151\, t^{\frac{1}{2}}$

TABLE VII

TEST 53

Plate thickness 1/4"

Time (sec)	Reference Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	81	85		81	
$\frac{1}{2}$	157	85		121	
1	207	87		175	
2	271	97		245	41.2
3	320	120		292	41.4
4	362	149		332	40.7
6	436	212		403	39.7
8	503	278		466	39.7
10	572	345		530	41.2
12	638	410		592	41.7
14	701	474		657	41.6
16	764	535		717	41.8
18	830	598		784	42.2
20	891	659		848	43.0
22	952	722		903	43.8
24	1009	786		965	44.5
26	1066	851		1016	45.1
28	1129	911		1072	46.1

Loss included with $q_\ell = 0.140\ t^{\frac{1}{2}}$

TABLE VIII

TEST 62

Plate thickness 1/4"
Slug thickness 1/4"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	69	67	
$\frac{1}{4}$	114	69	
$\frac{1}{2}$	154	77	
3/4	178	88	
1	202	100	45.5
$1\frac{1}{2}$	242	118	44.9
2	272	140	45.5
3	326	181	45.0
4	372	223	44.5
6	457	304	43.7
8	538	379	40.8
10	609	454	38.8
12	679	521	37.8
14	753	586	36.4
16	827	650	36.2
18	398	712	36.2
20	965	774	34.5
22	1028	832	33.9
24	1092	887	33.8
26	1154	945	33.0
28	1217	999	32.4
30	1273	1065	32.4

TABLE IX

TEST 63

Plate thickness 1/4"
Slug thickness 1/4"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	66	65	
$\frac{1}{4}$	119	67	
$\frac{1}{2}$	160	78	
3/4	190	89	
1	214	102	
$1\frac{1}{2}$	251	125	
2	280	144	43.[illegible]
3	332	186	42.4
4	378	222	41.0
6	457	292	41.0
8	530	370	40.3
10	601	442	39.4
12	672	510	38.5
14	744	578	37.4
16	815	644	36.7
18	887	706	35.8
20	953	767	35.2
22	1021	830	35.0
24	1088	885	34.4
26	1153	944	34.2
28	1215	1002	33.2
30	1280	1057	32.4
32	1348	1107	31.9

TABLE X

TEST 76

Plate thickness 1/8"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	67	67	
$\frac{1}{4}$	100	75	
$\frac{1}{2}$	120	88	
3/4	138	101	
1	151	113	
$1\frac{1}{2}$	176	135	24.74
2	201	158	25.44
$2\frac{1}{2}$	225	182	25.26
3	250	206	25.03
4	290	250	24.44
5	334	293	23.86
6	377	334	23.16
7	422	375	23.05
8	460	418	22.81
9	499	455	22.52
10	538	495	22.12
11	577	538	21.36
12	616	573	20.95
14	688	642	19.79
16	763	711	19.38
18	836	778	19.09
20	900	843	18.62
22	970	906	18.33
24	1035	967	18.04
26	1097	1026	17.75
28	1157	1083	17.46

TABLE XI

TEST 77

Plate thickness 1/8"
Slug thickness 1/4"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	68	67	
¼	102	71	
½	124	77	
3/4	141	84	
1	155	91	
1½	180	103	
2	203	114	
2½	228	126	
3	249	132	
4	296	161	25.25
5	341	185	25.86
6	384	208	"
7	428	231	"
8	470	255	"
9	508	279	"
10	551	305	"
12	629	345	"
14	705	397	"
16	778	442	"
18	852	488	"
20	921	532	"
22	988	578	"
24	1055	624	"
26	1120	669	"
28	1182	714	"

TABLE XII

TEST 78

Plate thickness 1/8"
Slug thickness 1/4"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	72	72	
1/4	105	74	
1/2	128	80	
3/4	145	88	
1	158	94	25.63
1 1/2	184	106	25.75
2	209	119	"
2 1/2	232	130	"
3	256	142	"
4	298	166	"
5	341	190	"
6	383	212	"
7	425	236	"
8	466	260	"
9	507	283	"
10	546	306	"
11	585	328	"
12	623	352	"
14	698	397	"
16	772	444	"
18	845	490	25.86
20	910	536	25 86
22	979	585	26.10
24	1042	628	26.45
26	1103	673	"
28	1162	717	"
30	1219	762	"

TABLE XIII

TEST 79

Plate thickness 1/8"

Time (sec)	Reference Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	70	70	70	70	
$\frac{1}{4}$	107	70	72	105	
$\frac{1}{2}$	132	71	82	128	
3/4	151	74	92	144	
1	166	79	102	159	21.7
$1\frac{1}{2}$	191	92	122	182	21.8
2	214	108	141	204	21.9
$2\frac{1}{2}$	239	124	159	225	21.7
3	259	140	178	244	"
4	303	173	214	285	"
5	345	205	251	325	"
6	387	237	288	362	21.9
7	428	270	324	397	21.7
8	468	303	360	432	21.5
9	502	335	397	470	21.3
10	540	367	431	503	21.1
11	584	400	463	531	"
12	621	428	497	564	"
14	692	488	560	628	"
16	764	548	625	688	"
18	835	605	688	749	"
20	905	663	750	807	21.0
22	970	718	810	867	21.0
24	1036	775	868	922	20.6
26	1098	828	927	978	20.5
28	1158	883	984	1031	20.3
30	1215	938	1040	1083	19.8
32	1268	990	1092	1130	19.6

Loss included with $q_{\ell} = 0.135\ t^{\frac{1}{2}}$

TABLE XIV

TEST 82

Plate thickness 1/8"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	85	86	
¼	118	96	
½	140	107	
3/4	156	118	
1	170	128	
1½	194	149	
2	217	170	24.4
2½	237	192	24.4
3	260	214	24.1
4	304	259	24.0
5	344	302	23.3
6	386	342	22.8
7	428	380	22.7
8	467	422	22.4
9	506	460	22.2
10	550	499	21.8
11	588	538	21.7
12	625	576	21.5
14	700	651	20.8
16	773	721	20.0
18	842	788	19.3
20	913	854	18.9
22	979	918	18.7
24	1044	980	18.2

TABLE XV

TEST 83

Plate thickness 1/8"

Time (sec)	Reference Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	79	78	78	79	
$\frac{1}{4}$	90	78	80	91	
$\frac{1}{2}$	110	78	84	112	
3/4	123	80	91	125	18.0
1	133	84	100	136	18.3
$1\frac{1}{2}$	150	94	116	154	17.2
2	164	105	132	172	16.8
$2\frac{1}{2}$	181	118	148	188	16.6
3	195	130	162	202	16.8
4	222	156	192	233	16.9
5	250	181	221	262	17.0
6	278	207	249	294	"
7	306	233	278	322	"
8	332	258	306	350	"
9	359	284	334	378	"
10	386	310	361	406	"
12	438	359	416	460	17.1
14	488	409	469	512	17.2
16	539	455	524	570	17.2
18	589	506	574	624	17.1
20	637	552	625	673	16.9
22	684	600	673	724	16.9
24	732	645	722	772	16.8
26	777	692	773	822	"
28	823	737	818	870	"
30	866	782	865	915	"
34	954	875	958	1005	"
38	1038	964	1048	1096	"

Loss included with $q_\ell = 0.110\, t^{\frac{1}{2}}$

TABLE XVI

TEST 85

Plate thickness 1/8"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	75	75	
$\frac{1}{4}$	92	79	
$\frac{1}{2}$	107	88	
3/4	118	98	22.0
1	128	108	22.4
$1\frac{1}{2}$	145	128	20.9
2	160	147	20.9
$2\frac{1}{2}$	175	167	21.4
3	190	186	21.4
4	220	225	21.4
5	246	264	20.9
6	274	300	20.7
7	302	337	20.1
8	328	374	20.1
10	382	442	19.7
12	435	510	18.9
14	489	577	18.8
16	537	642	18.0
18	585	705	17.6
20	634	767	17.3
22	682	822	16.9
24	731	881	16.8
26	775	936	16.3
30	863	1041	16.0
34	952	1151	16.0
38	1032	1263	16.0

TABLE XVII

TEST 90

Plate thickness 1/8"

Time (sec)	Reference Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	76	73	74	74	
¼	95	73	75	96	23.2
½	118	74	84	121	22.4
3/4	134	77	94	138	22.1
1	147	82	104	150	22.2
1½	168	95	125	173	22.4
2	188	111	145	193	21.8
2½	207	127	165	213	21.4
3	225	144	184	231	21.4
4	263			270	21.5
5	298	210	257	308	21.5
6	332	242	293	343	21.5
7	365	275	329	379	21.6
8	400	307	363	414	21.8
10	464	372	435	483	22.2
12	533	431	508	557	22.3
14	599	495	577	626	22.4
16	665	556	645	692	22.6
18	726	618	710	761	22.6
20	787	680	773	825	22.7
22	849	743	840	890	22.8
24	911	804	905	953	23.0
26	970	865	970	1022	23.1
28	1026	926	1031	1080	23.1
30	1088	984	1092	1143	22.9

Loss included with $q_\ell = 0.140\, t^{\frac{1}{2}}$

TABLE XVIII

TEST 91

Plate thickness 1/8"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	80	80	
¼	102	86	
½	120	97	26.3
3/4	135	109	26.3
1	147	121	25.9
1½	168	145	26.1
2	187	170	25.8
2½	205	194	25.6
3	222	217	25.4
4	258	263	"
5	293	309	"
6	328	355	"
7	363	400	24.8
8	398	443	24.5
10	464	526	24.1
12	530	615	24.0
14	593	699	23.9
16	654	778	23.3
18	718	857	23.0
20	780	935	22.7
22	840	1009	22.4
24	899	1082	"
26	956	1155	"
28	1015	1230	"
30	1070	1294	"

TABLE XIX

TEST 92

Plate thickness 1/8"
Slug thickness 1/8"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	73	72	
¼	95	79	
½	116	90	26.7
3/4	130	102	26.2
1	143	113	26.1
1½	164	138	25.6
2	183	161	24.9
2½	202	184	24.3
3	222	206	24.2
4	258	248	23.3
5	296	292	23.1
6	332	333	22.6
7	368	372	22.3
8	405	410	22.1
10	474	490	22.0
12	547	569	22.0
14	616	647	21.7
16	678	720	21.4
18	746	793	21.3
20	809	863	20.7
22	870	932	20.4
24	935	1001	20.2

TABLE XX

TEST 99

Plate thickness 1/4"
Slug thickness 1/4"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	82	83	
$\frac{1}{4}$	130	87	
$\frac{1}{2}$	149	93	
3/4	163	99	25.4
1	177	105	25.4
$1\frac{1}{2}$	196	117	25.6
2	214	128	25.6
$2\frac{1}{2}$	228	139	25.4
3	242	151	25.2
4	267	174	25.0
5	289	198	25.0
6	311	222	25.0
8	356	265	24.5
10	400	311	23.8
12	440	351	23.1
14	481	392	23.1
16	522	432	23.1
18	574	472	22.7
20	612	516	22.4
24	692	592	21.6
28	769	665	21.0
32	847	736	20.4
36	922	806	20.2
40	994	874	19.8
44	1064	938	19.7

TABLE XXI

TEST 101

Plate thickness 1/8"

Time (sec)	Reference Temperature (°F)	Back Temperature (°F)	Midplane Temperature (°F)	Surface Temperature (°F)	Heating Rate (Btu/ft^2-sec)
0	80	79	81	80	
$\frac{1}{4}$	100	79	82	105	
$\frac{1}{2}$	125	79	92	132	
3/4	142	82	103	150	22.5
1	154	87	113	162	22.5
$1\frac{1}{2}$	175	102	135	185	21.7
2	195	117	155	205	21.8
$2\frac{1}{2}$	212	129	174	226	21.9
3	231	152	194	245	21.9
4	266	185	232	283	21.9
5	299	218	269	320	21.8
6	332	251	304	355	21.7
7	365	284	340	392	21.7
8	396	316	375	426	21.7
10	460	380	446	495	21.9
12	530	444	518	560	22.0
14	589	505	585	627	22.0
16	650	567	650	691	22.1
18	713	628	716	755	22.2
20	771	692	780	818	22.3
22	832	751	843	884	22.5
24	887	810	906	946	22.6
28	999	932	1028	1064	22.4
32	1112	1047	1145	1182	22.4

Loss included with $q_{\ell} = 0.140\ t^{\frac{1}{2}}$

TABLE XXIII

TEST 107

Plate thickness 1/8"
Slug thickness 1/4"

Time (sec)	Reference Temperature (°F)	Slug Temperature (°F)	Indicated Heating Rate (Btu/ft^2-sec)
0	82	81	
¼		85	
½	125	92	27.5
3/4		98	27.3
1	152	105	27.3
1½	172	118	27.3
2	192	130	28.1
2½	209	143	28.1
3	227	155	28.0
4	262	180	27.6
5	299	207	27.4
6	334	232	27.5
7	370	256	"
8	402	281	"
10	470	332	"
12	534	380	"
14	598	431	"
16	662	480	"
18	724	528	"
20	786	578	"
22	845	624	"
24	905	674	"
28	1019	768	"
32	1127	862	"

APPROVAL

EVALUATION OF HEAT FLUX METERS

Phase 2 - Experimental Investigation

The information in this report has been reviewed for security classification. Review of any information concerning Department of Defense or Atomic Energy Commission programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

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